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Braud

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(45) Date of Patent: Dec. 2, 2003

(54) GEARBOX, PARTICULARLY FOR AUTOMOTIVE VEHICLE WITH TELESCOPIC LOAD-CARRYING ARM

(75) Inventor: Marcel-Claude Braud, Saint Herblon (FR)

(73) Assignee: Manitou BF, Ancenis (FR)

(*) Notice: Subject to any disclaimer, the term of this patent is extended or adjusted under 35 U.S.C. 154(b) by 74 days.

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(51) Int. Cl.[7] B60K 17/34

(52) U.S. Cl. .... 180/233; 180/374; 180/384; 74/665 GB

(58) Field of Search .... 180/233, 384, 180/364, 365, 374; 74/664, 665 R, 321, 665 GB, 665 GC, 665 H

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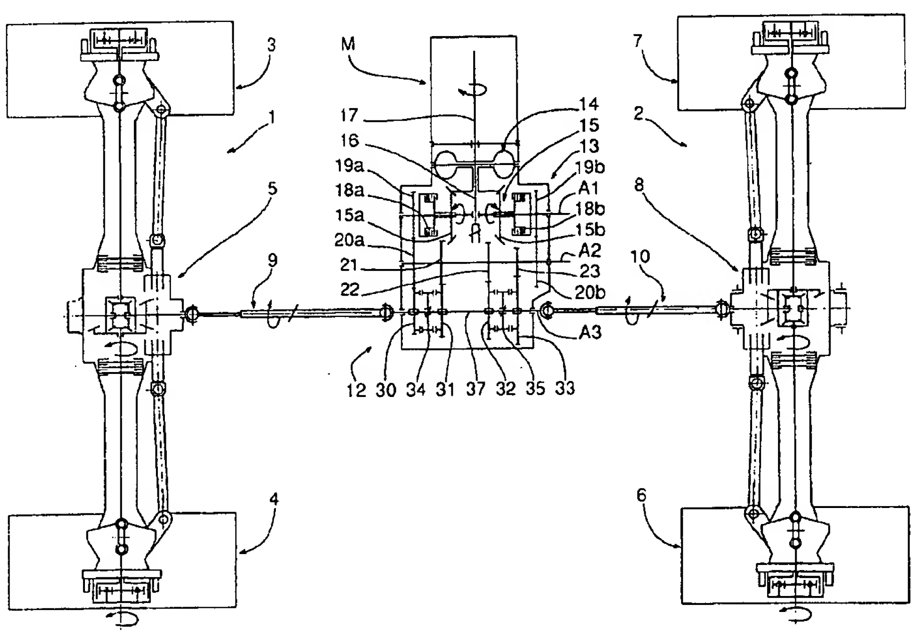
Primary Examiner—Anne Marie Boehler
(74) *Attorney, Agent, or Firm*—Young & Thompson

(57) **ABSTRACT**

A gearbox, particularly for an automotive vehicle with a telescopic load-carrying arm, is adapted to transmit directly or indirectly the drive movement of an internal combustion motor (M) with at least one front or rear axle provided with driven wheels. This gearbox includes a casing (13) containing an angled transmission (15) arranged to transmit the movement of the output shaft (17) of the motor, to a drive shaft (9 or 10) of the front or rear axle, or both. The gearbox further comprises a power takeoff through shaft (16) arranged to drive a pump that generates hydraulic or hydrostatic energy.

8 Claims, 6 Drawing Sheets



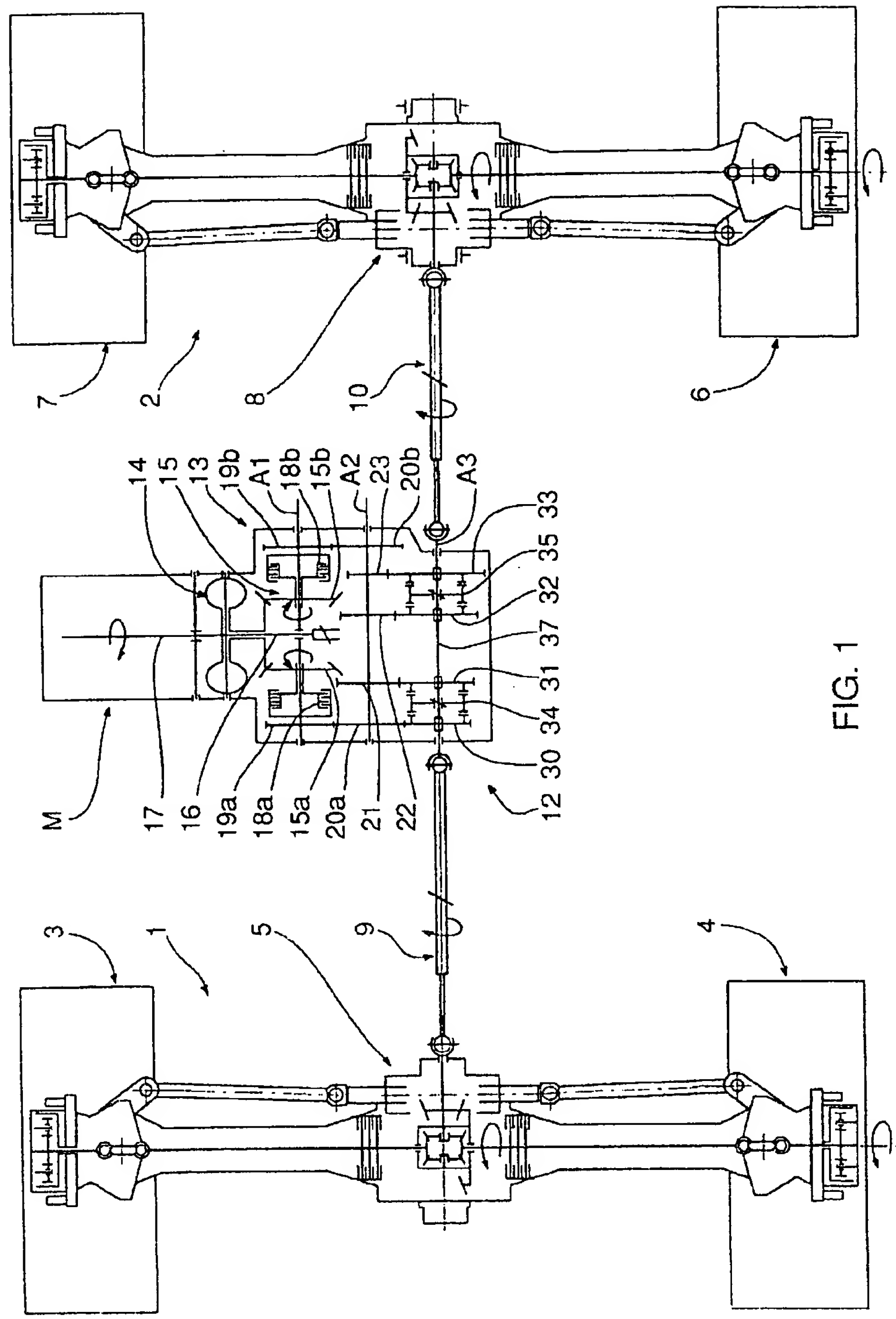


FIG. 1

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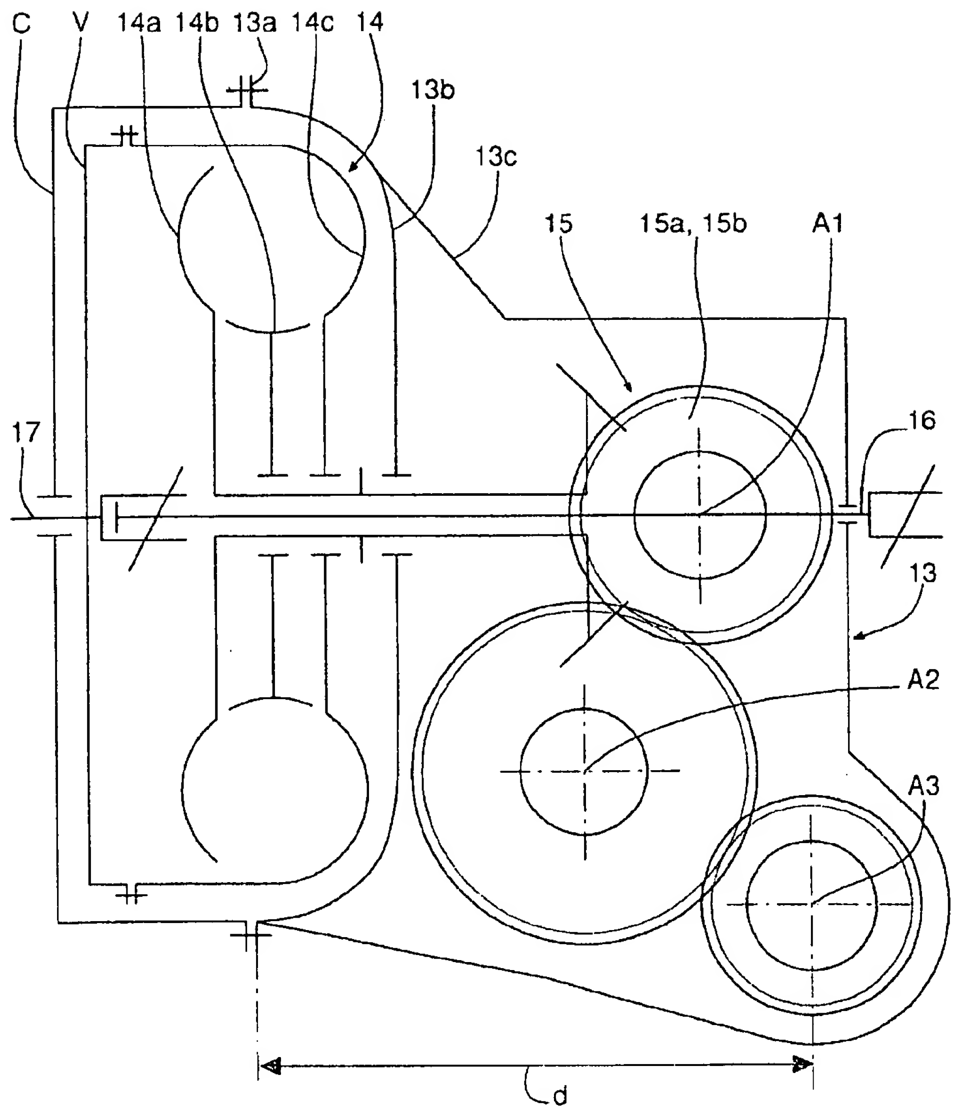

FIG. 2

	18a	18b	30	31	32	33
1ère AV	1	0	0	0	0	1
2ème AV	1	0	0	0	1	0
3ème AV	1	0	0	1	0	0
4ème AV	1	0	1	0	0	0
1ère AR	0	1	0	0	0	1
2ème AR	0	1	0	0	1	0
3ème AR	0	1	0	1	0	0
4ème AR	0	1	1	0	0	0

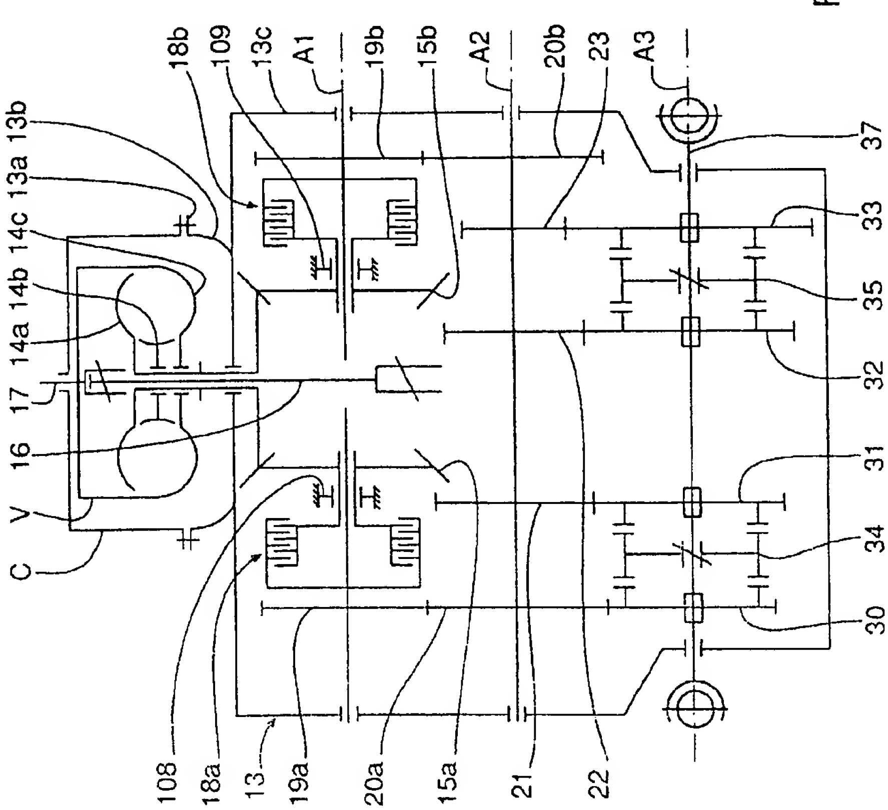

FIG. 3

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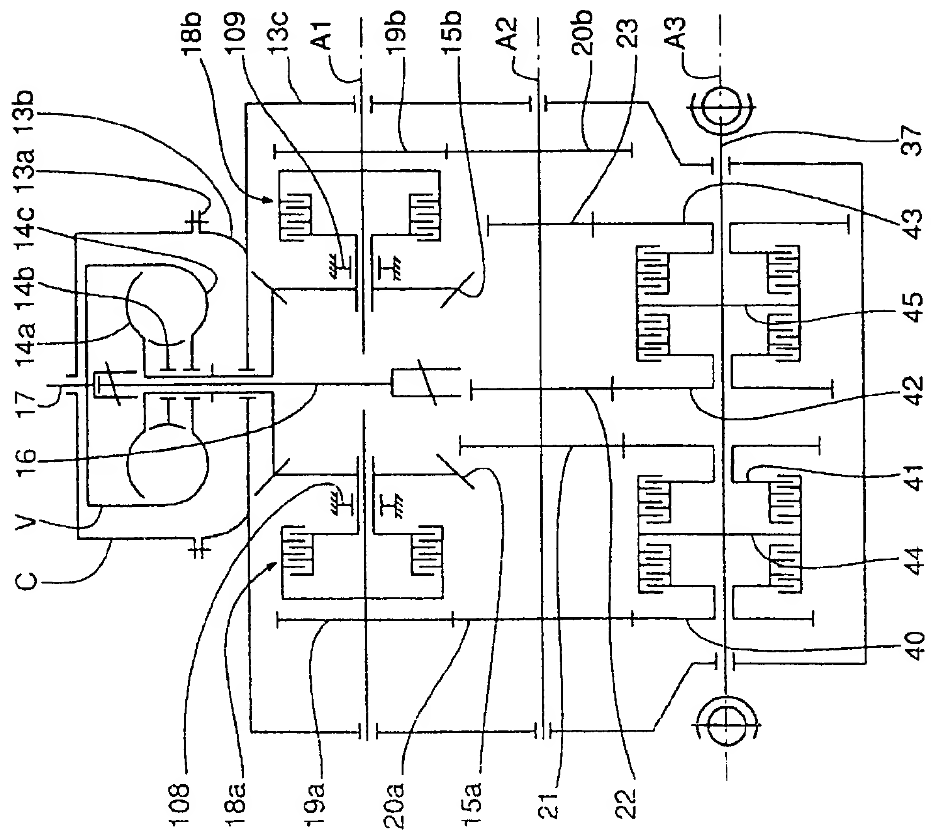
	18a	18b	40	41	42	43
1ère AV	1	0	0	0	0	1
2ème AV	1	0	0	0	1	0
3ème AV	1	0	0	1	0	0
4ème AV	1	0	1	0	0	0
1ère AR	0	1	0	0	0	1
2ème AR	0	1	0	0	1	0
3ème AR	0	1	0	1	0	0
4ème AR	0	1	1	0	0	0

FIG. 4



	103	104	107	52	51	53
1ère AV	0	1	0	1	0	0
2ème AV	1	0	0	1	0	0
3ème AV	0	1	0	0	1	0
4ème AV	1	0	0	0	1	0
5ème AV	0	1	0	0	0	1
6ème AV	1	0	0	0	0	1
1ère AR	0	0	1	1	0	0
2ème AR	0	0	1	0	1	0
3ème AR	0	0	1	0	0	1

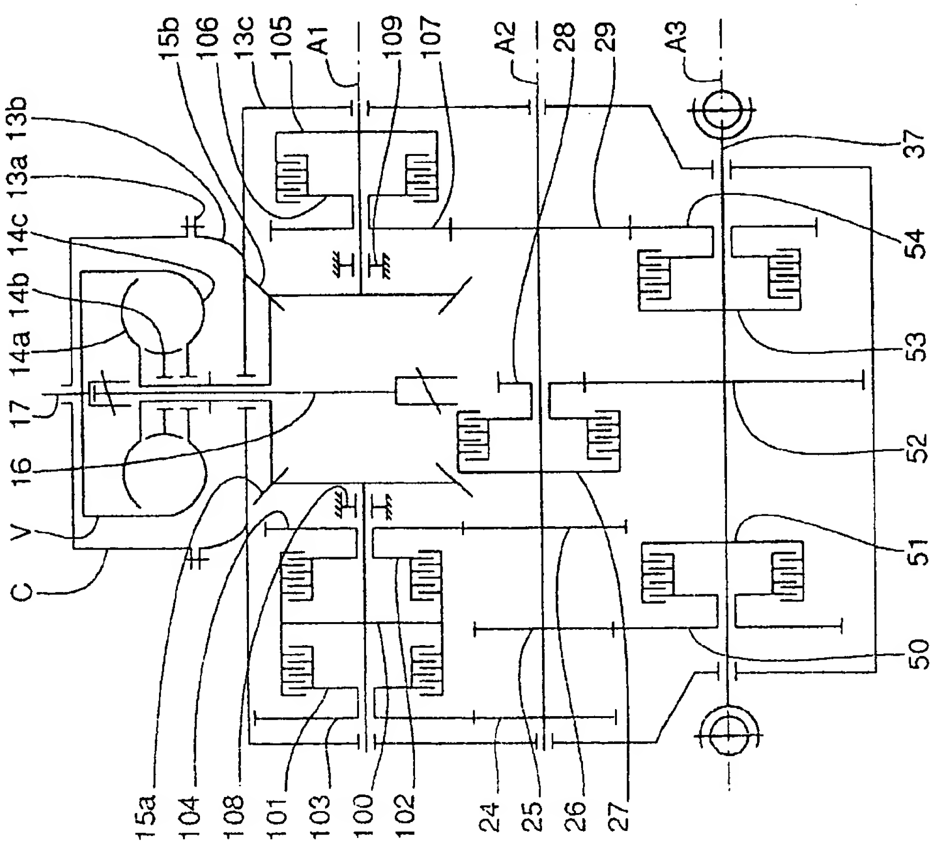


FIG. 5

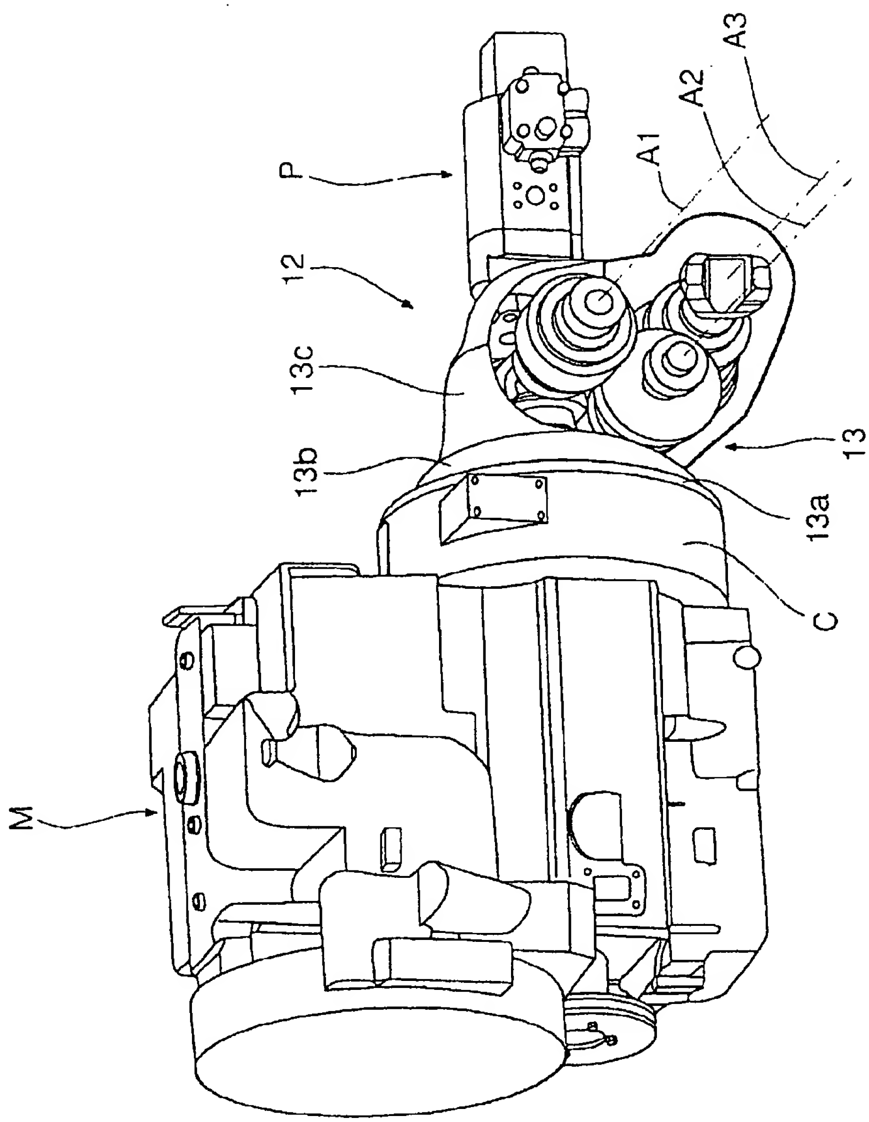


FIG. 6

GEARBOX, PARTICULARLY FOR AUTOMOTIVE VEHICLE WITH TELESCOPIC LOAD-CARRYING ARM

CROSS REFERENCE TO RELATED APPLICATION

This application is based on the disclosure of French application 00.12861 filed in France on Oct. 9, 2000, the entirety of which application is hereby expressly incorporated by reference.

FIELD OF THE INVENTION

The invention relates to gearboxes, particularly for automotive vehicles with telescopic load-bearing arms.

The invention also relates to an automotive vehicle with a telescopic load-bearing arm provided with a gearbox according to the invention.

BACKGROUND OF THE INVENTION

Automotive vehicles with telescopic load-bearing arms are known, produced and sold by the MANITOU BF Company in France, in which a diesel motor is oriented transversely to the longitudinal axis of the machine.

The diesel motor drives an angled transmission, of which an output drives a gearbox oriented longitudinally and disposed substantially in a central position and of which another output continuously drives a hydraulic pump upon startup of the motor.

These machines are generally satisfactory, but have a large number of mechanical members (resilient coupling, cardan transmission, angled transmission, torque converter, gearbox), of which certain ones are mounted in individual casings before being assembled together.

This technique results in a substantial size of the assembly produced and a costly fabrication of the mechanical transmission means of these machines.

SUMMARY OF THE INVENTION

The invention has for its object to overcome the drawbacks of the prior art, by providing a new gearbox, adapted particularly for new automotive vehicles with a telescopic load-carrying arm, of simple and economical construction and of reduced size.

The invention has for its object a gearbox, particularly for an automotive vehicle with a telescopic load-carrying arm, adapted to transmit directly or indirectly the drive movement of a transversely oriented internal combustion engine, to at least one front or rear axle having drive wheels, by means of at least one longitudinally oriented shaft, characterized in that the casing of the gearbox contains an angled transmission, whose transversely oriented input is adapted to be driven by a clutch means, and whose longitudinally oriented output defines a first shaft line of the gearbox.

According to other advantageous characteristics of the invention:

- the gearbox comprises moreover a through power takeoff shaft arranged to drive a pump generating hydraulic or hydrostatic energy
- the angled transmission comprises a double conical pinion: a forward drive pinion and a rear drive pinion, with at least one associated clutch,
- the gearbox is driven by a torque converter connected to the output of the internal combustion motor by constituting the first drive member from the gearbox, whilst the angled transmission is the second driving member secured to the turbine of the converter.
- the casing of the gearbox contains at least three parallel shaft lines: a first shaft line corresponding to the angled transmission, a second intermediate shaft for transmission between the first shaft line and a third shaft line, this third shaft line corresponding to the shaft (or shafts) for driving the front or rear axle (or front or rear drive axles),
- the first shaft line corresponding to the angled transmission comprises a double conical pinion engaging with a conical pinion adapted to be driven by the output shaft of the motor, and a double clutch for selectively driving a toothed forward drive wheel or a toothed rearward drive wheel, and these two toothed front and rear drive wheels engaging continuously with toothed wheels of the second intermediate transmission shaft,
- the second shaft line comprises several toothed wheels for transmission of movement imparted by the first shaft line to the corresponding toothed wheels of the third shaft line with which they are continuously in engagement,
- the corresponding toothed wheels of the third shaft line are mounted freely in rotation on the third shaft, and are adapted to drive this third shaft under the action of clutch or toothed means,
- the third shaft line moreover comprises a toothed wheel mounted securely on the third shaft and engaging with a toothed wheel disengageable from the second shaft line, so as to provide a number of forward drive speeds that is greater than the number of rearward drive speeds,
- the third shaft line comprises two shaft output adapted to drive simultaneously a front drive axle and a rear drive axle of an automotive vehicle, particularly a vehicle with a telescopic load-carrying arm.

The invention also has for its object an automotive vehicle with a telescopic load-carrying arm, of the type comprising two front and rear axles provided with wheels, and an internal combustion engine driving a gearbox.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will be better understood from the description which follows, given by way of non-limiting example, with reference to the accompanying drawings, in which:

FIG. **1** shows schematically in a view from above the kinematic arrangement of one embodiment of automotive vehicle according to the invention.

FIG. **2** shows schematically a side view of a gearbox according to the invention.

FIG. **3** shows schematically a kinematic chain of a first embodiment of gearbox according to the invention.

FIG. **4** shows schematically a kinematic chain of a second embodiment of gearbox according to the invention.

FIG. **5** represents schematically a kinematic chain of a third embodiment of gearbox according to the invention.

FIG. **6** shows schematically a perspective view of an assembly comprising a gearbox according to the invention shown with the casing partially broken away.

DETAILED DESCRIPTION OF THE INVENTION

Referring to FIG. **1**, an embodiment of a vehicle according to the invention comprises a kinematic arrangement with

a front axle **1** and a rear axle **2**. Front axle **1** is a self-blocking differential axle with limited sliding, driving two front driven and steering wheels **3** and **4**. The control of the front axle is a hydrostatic control with a central double-acting jack **5**.

The rear axle **2** is a self-blocking differential axle with limited sliding to drive and orient two driven and steerable rear wheels **6** and **7**.

The control of the rear axle is also a hydrostatic control with a central double-acting jack **8**.

The drive of the front axle **1** is ensured by means of a cardan shaft **9**, whilst the drive of the rear axle **2** is ensured by means of a cardan shaft **10**. The cardan shafts **9** and **10** are connected to two corresponding power outputs of a gearbox **12** designed such that the cardan shafts **9** and **10** will be driven simultaneously at the same speed when the vehicle moves and such that the cardan shafts **9** and **10** will be simultaneously stopped when the vehicle is stopped.

The gearbox **12** comprises a casing **13** surrounding a torque converter **14** or equivalent clutch with a hollow shaft, adapted to drive an angled transmission **15** with a double conical pinion.

The converter **14** with a hollow shaft and the angled transmission **15** are traversed by a central shaft **16** for power takeoff, secured to the output shaft **17** of the motor M, or to the inertial flywheel of the internal combustion engine M so as continuously to drive the power takeoff shaft **16** upon starting up the motor M. The power takeoff shaft **16** is preferably adapted to drive a pump generating hydraulic or hydrostatic energy, not shown and external to the casing **13** of the gearbox **12**.

Preferably, the converter **14** with a hollow shaft or equivalent clutch is connected to the output drive of the internal combustion motor M and constitutes the first member driving the gearbox **12**, whilst the angled transmission **15** is the second driving member secured to the turbine of the converter **14** with a hollow shaft and integrated into the casing **13** of the gearbox **12**.

The angled transmission **15** with double conical pinion comprises a forward drive pinion **15*a*** with an associated clutch **18*a*** and a reverse drive pinion **15*b*** with an associated clutch **18*b***.

The casing **13** of the gearbox **12** contains three parallel shaft lines **A1**, **A2**, **A3**.

The first shaft line A1 corresponding to the angled transmission **15** comprises for forward drive: the pinion **15*a***, the associated clutch **18*a*** and the toothed forward drive wheel **19*a***; and for reverse drive: the pinion **15*b***, the associated clutch **18*b*** and the reverse drive toothed wheel **19*b***.

The toothed wheels for forward drive **19*a*** or reverse drive **19*b*** continuously engage with the toothed wheels **20*a***, **20*b*** of the second intermediate shaft A2 of the transmission, which also comprises toothed wheels **21**, **22**, **23** fixed in rotation with the toothed wheels **20*a***, **20*b***.

Each toothed wheel **20*a*** or **21** or **22** or **23** continuously engages with the toothed wheel **30** or **31** or **32** or **33** of the third shaft line A3 for driving the cardan shafts **9** and **10** of front and rear axles **1** and **2**.

The toothed wheels **30** to **33** are mounted freely in rotation on the third shaft line with the possibility of engagement or equivalent synchronous drive with the crowns **34**, **35** mounted securely in rotation with the third shaft line, preferably mounted on grooves on the third shaft **37** with the possibility of lateral movement under the action of mechanical controls of a type known per se and not illustrated.

The toothed engagement of a crown **34** or **35** with one of the toothed wheels **30** to **33** thus permits selecting the forward speed or rearward speed.

With reference to FIGS. **2** to **4**, identical or functionally equivalent elements to FIG. **1** are indicated by the same reference numerals.

In FIG. **2**, the casing **13** comprises flange **13*a*** for mounting on the casing C of the flywheel V of the motor M (not shown). The converter **14** with a hollow shaft comprises a turbine **14*a*** with a hollow shaft for driving the angled transmission **15**, a reactor **14*b***, and an impeller **14*c*** secured to the flywheel V of the motor M (not shown). The power takeoff shaft **16** is driven by the output shaft **17** of the motor M and passes between the pinions **15*a***, **15*b*** to leave the casing **13** and to drive a hydraulic pump (not shown).

The three shaft lines described with reference to FIGS. **1** to **6** are schematically shown by the geometrical axes A1, A2, A3.

The reduced size of the casing **13** and of the gearbox **12** according to the invention corresponds to the mounting of the three shaft lines A1, A2, A3 on sealed bearings of known type and does not require a more detailed description.

The reduction of size in the transverse direction permits obtaining a reduced distance d between the flange **13*a*** and the axis A3 of the shaft **37**, preferably comprised between 200 and 500 mm and permits mounting substantially in alignment the cardan shafts **9** and **10** and the third shaft **37**, thereby avoiding the addition of any intermediate transmission and any supplemental gearing.

The shaft **37** is thus preferably located, after mounting, substantially in the median longitudinal plane of the machine for which the gearbox **12** is provided.

FIG. **3** shows schematically a first embodiment of gearbox according to the invention adapted for an automotive vehicle of the type described with reference to FIG. **1**, comprising reference numerals identical to the reference numerals of FIGS. **1** and **2**.

In this embodiment, the casing **13** of gearbox comprises a bell **13*b*** surrounding the torque converter **14** or equivalent clutch, and a sealed chamber **13*c*** receiving a hydraulic liquid for lubricating the gearing and the clutches of the gearbox **12**.

Bell **13*b*** of casing **13** of gearbox **12** is mounted on the casing C of the motor flywheel V by its flange **13*a***.

The selection of the speed ratio results from the coaction or the reciprocal activity of the different clutch or mechanically toothed members, as indicated in the speed table in which the numeral **1** indicates the clutching activity **18*a*** or **18*b***, or the drive of a crown **34** or **35** and the drive axles **1** and **2** by one of the toothed wheels **30** to **33** of the shaft **34**, whilst the numeral **0** indicates the dead point of the clutch **18*a*** or **18*b*** or the absence of activity of a toothed wheel **30** to **33** in the free wheel condition on shaft **37**.

FIG. **4** shows schematically a second embodiment of gearbox according to the invention, comprising reference numerals identical to those of FIG. **2**.

This second embodiment differs from the first embodiment by the mechanical arrangement of the third shaft line A3, which comprises double clutches, permitting speed change under torque, the "power shift" transmission so called by specialists, with electrical or hydraulic control of known type and not shown in detail.

In the first and second embodiments described with reference to FIGS. **3** and **4**, it is possible to select four forward speeds and four reverse speeds, to turn the shaft **37**

in the corresponding regime and in the corresponding direction and to transmit its movement by cardan shafts **9** and **10** in the middle of the drive axles **1** and **2**.

The selection of the speed ratio results from the coaction or reciprocal inactivity of the different mechanical clutch members, as indicated in the speed table, in which the numeral **1** indicates the clutching activity **18***a* or **18***b* or the drive by one of the clutch wheels **40** to **43** with a plate **44** or **45** of the shaft **37** and of the drive axles **1** and **2**, whilst the numeral **0** indicates the dead point of the clutch **18***a* or **18***b*, or the absence of clutching of a toothed wheel **40** to **43** in the free wheel condition on shaft **37**.

With reference to FIG. **5**, identical or functionally equivalent elements to those of FIG. **4** are shown by the same reference numerals.

In FIG. **5**, a third embodiment of gearbox according to the invention permits changing speeds under torque, the "power shift" transmission so called by specialists, by modifying the mechanical arrangements of the three shaft lines **A1**, **A2**, **A3**. These arrangements comprise for example intermediate support bearings **108** and **109** for the first shaft line **A1** comprising the double conical pinion **15***a* or **15***b*, a double hydraulic forward drive clutch with three plates **100–101–102**, and associated toothed wheels **103** and **104** secured respectively to the plates **101** and **102**, and a hydraulic clutch with two plates **105–106** for reverse drive with an associated toothed wheel **107** secured to the plate **106**. The toothed wheels **103**, **104** and **107** of the first shaft line **A1** engage continuously with the corresponding toothed wheels **24**, **26** and **29** of the second shaft line **A2**.

A wheel **25** or **28** or **29** of the second shaft line **A2** can transmit the movement of the second shaft line **A2** to the third shaft **37**, as a function of the force of the various clutches of the gearbox, and in particular the clutch **27–28** of the second shaft line **A2**.

The toothed wheels **25**, **28** and **29** engage with toothed wheels **50**, **52** and **54** of the third shaft **37**: the first and third toothed wheels **50** and **54** are mounted disengageably and engageably under the action of clutch means **50–51** and **53–54** respectively. The second toothed wheel **52** is mounted fixedly on the shaft **37**, because the toothed wheel **28** of the second shaft line **A2** is disengageable.

These arrangements permit obtaining six forward speeds and three reverse speeds, as will be seen from the speed table indicating the condition of driving (**1**) or not (**0**) of one of the toothed wheels **103**, **104** or **107** or of a member **51**, **52** or **53** of the third shaft **37** as a function of the force of the various forward drive clutches **100–101**, **100–102**, or of reverse drive **105–106** and selection clutches **50–51**, **27–28**, **53–54** of the two shaft lines **A2** and **A3**.

With reference to FIG. **6**, an arrangement with three parallel shaft lines **A1**, **A2**, **A3** of a gearbox according to the invention provides the advantage of having shaft outputs to drive the front and rear axles substantially in the desired plane (in general the longitudinal median plane of the corresponding machine), and moreover permits substantially reducing the longitudinal size of the mechanical assembly comprising the motor M and the gearbox **12**.

Similarly, the passage provided through the hollow shaft of the converter **14** to transmit the movement of drive shaft **17** permits reducing the transverse size of the mounting of the hydraulic pumps P driven by the power takeoff shaft **16**, such that the assembly shown comprising the motor M, the gearbox **12** and these hydraulic pumps P has a limited transverse size.

Moreover, the use of angled transmission **15** with a double conical pinion **15***a*, **15***b* supplying alternatively forward movement and reverse movement, permits having a relay shaft to reverse the direction of rotation for reverse drive.

The invention described with reference to particular embodiments is in no way thereby limited, but on the contrary covers any modification of form and all variation of embodiment within the scope and spirit of the invention.

Thus, the invention also covers any modified embodiment not shown, of a gearbox comprising at least three parallel shaft lines and arranged substantially longitudinally, having a reduced size, and of simple and economic construction. To this end, there can be used an angled transmission with a single conical pinion instead of an angled transmission with a double conical pinion described with reference to FIGS. **1** to **6**, so as to reduce the longitudinal size of the gearbox by a length corresponding to the reverse drive pinion **15***b*, to the associated clutch **18***b* and to the toothed reverse drive wheel **19***b*. This modification to reduce the size requires providing a fourth shaft, namely a reversing shaft for reverse movement, parallel to the three shaft lines **A1**, **A2**, **A3** described with reference to FIGS. **1** to **6**.

This reversing shaft for reverse drive coacts disengageably with the shafts **A1** and **A2**; when this reversing shaft is disengaged, the gear box according to the invention operates in forward drive; when this reversing shaft is engaged, the movement of shaft **A1** drives the reversing shaft which in turn drives the shaft **A2** and the gearbox operates in reverse drive.

The increase of the dimension of the gearbox in the transverse direction by virtue of the addition of this reversing shaft is compensated by the decrease of the longitudinal size of the gearbox, such that the volume of the casing of this modification is comparable to the volume of the casing **13** described with reference to FIGS. **5** and **6**.

What is claimed is:

1. A gearbox adapted to transmit drive movement from an internal combustion motor oriented transversely, to at least one front or rear axle provided with driven wheels via at least one longitudinally oriented shaft, the gearbox comprising:

a casing that contains an angled transmission having a transversely oriented input and a longitudinally oriented output defining a first shaft line of the gearbox;

at least three parallel shaft lines comprising the first shaft line corresponding to the angled transmission, a second shaft line for transmission between the first and a third shaft line, said third shaft line corresponding to a drive shaft of a said front or rear axle;

the first shaft line corresponding to the angled transmission comprising a double conical pinion engaging with a conical portion adapted to be driven by an output shaft of the motor and a double clutch for selectively driving a toothed forward drive wheel or a toothed reverse drive wheel, said two toothed wheels engaging continuously with toothed wheels on said second intermediate transmission shaft; and

a torque converter structured and arranged to drive the transversely oriented input.

2. The gearbox according to claim **1**, further comprising a power takeoff through shaft that drives a pump which generates hydraulic or hydrostatic energy.

3. The gearbox according to claim **1**, wherein said double conical pinion comprises a forward drive pinion and a rearward drive pinion, with at least one associated clutch.

4. The gearbox according to claim **1**, wherein the second shaft line comprises several toothed wheels for transmission of movement imparted by the first shaft line to correspond-

ing toothed wheels of the third shaft line with which said toothed wheels are continuously in engagement.

5. The gearbox according to claim **4**, wherein said corresponding toothed wheels of the third shaft line are mounted freely in rotation on said third shaft, and are adapted to drive said third shaft under the action of clutch or toothed means.

6. The gearbox according to claim **5**, wherein the third shaft line further comprises a toothed wheel mounted securely to the third shaft and engaging with a disengageable toothed wheel of the second shaft line, so as to provide a number of forward speeds greater than the number of reverse speeds.

7. The gearbox according to claim **6**, wherein the third shaft line comprises two output shafts adapted simultaneously to drive said front and rear axles.

8. An automotive vehicle comprising a telescopic load-carrying arm, and a gearbox as claimed in claim **1**.

* * * * *

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United States Patent [19]

Lanzer

[11] **Patent Number: 4,747,643**

[45] **Date f Patent: May 31, 1988**

[54] **MOTOR VEHICLE HAVING TWO DRIVEN AXLES AND BRAKE SYSTEM**

[75] Inventor: **Heribert Lanzer**, Gössendorf, Austria

[73] Assignee: **Steyr-Daimler-Puch Aktiengesellschaft**, Vienna, Austria

[21] Appl. No.: **42,355**

[22] Filed: **Apr. 24, 1987**

[30] **Foreign Application Priority Data**

Apr. 29, 1986 [AT] Austria A1140/86

[51] **Int. Cl.⁴** **B60K 17/34; B60T 8/32**

[52] **U.S. Cl.** .... **303/2**; 180/233; 180/248; 180/249; 188/181 T; 188/349; 303/9.62; 303/9.71; 303/100; 303/112; 303/113

[58] **Field of Search** .... 180/233, 248, 249, 197; 303/100, 20, 112, 111, 113, 110, 6; 188/181 T, 349, 181 A

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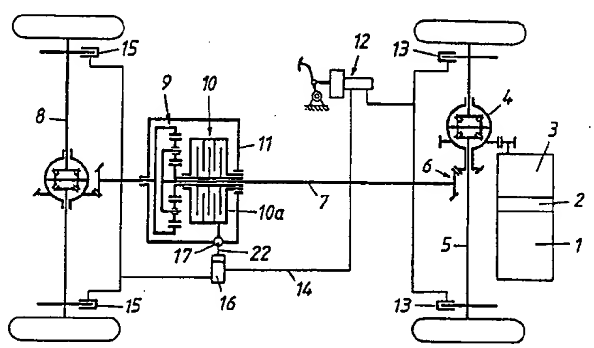
Primary Examiner—Douglas C. Butler
Attorney, Agent, or Firm—Marmorek, Guttman & Rubenstein

[57] **ABSTRACT**

A motor vehicle comprises two driven axles (5, 8), which are interconnected by an interaxle coupling (10) for transmitting a torque only when the halves of the coupling (10) rotate at different speeds and by a speed-changing transmission (9) connected in series with the coupling (10). A brake system (12 to 15) is provided for braking the wheels of both axles (5, 8). In order to ensure that the wheels of the driven rear axle (8) will not block before the wheels of the driven front axle (5), a pressure-setting controller (17) is adapted to control a brake pressure regulator (16) associated with the driven rear axle (8) and is responsive to a reversal of the direction in which torque is transmitted to and from the driven rear axle (8). During a transmission of torque from the driven rear axle (8) to the driven front axle (5), the pressure-setting controller (17) substantially reduces in dependence on the value of the torque being transmitted the pressure which is applied to brake the wheels of the driven rear axle (8) relative to the pressure which is applied to brake the wheels of the driven front axle (5).

7 Claims, 3 Drawing Sheets



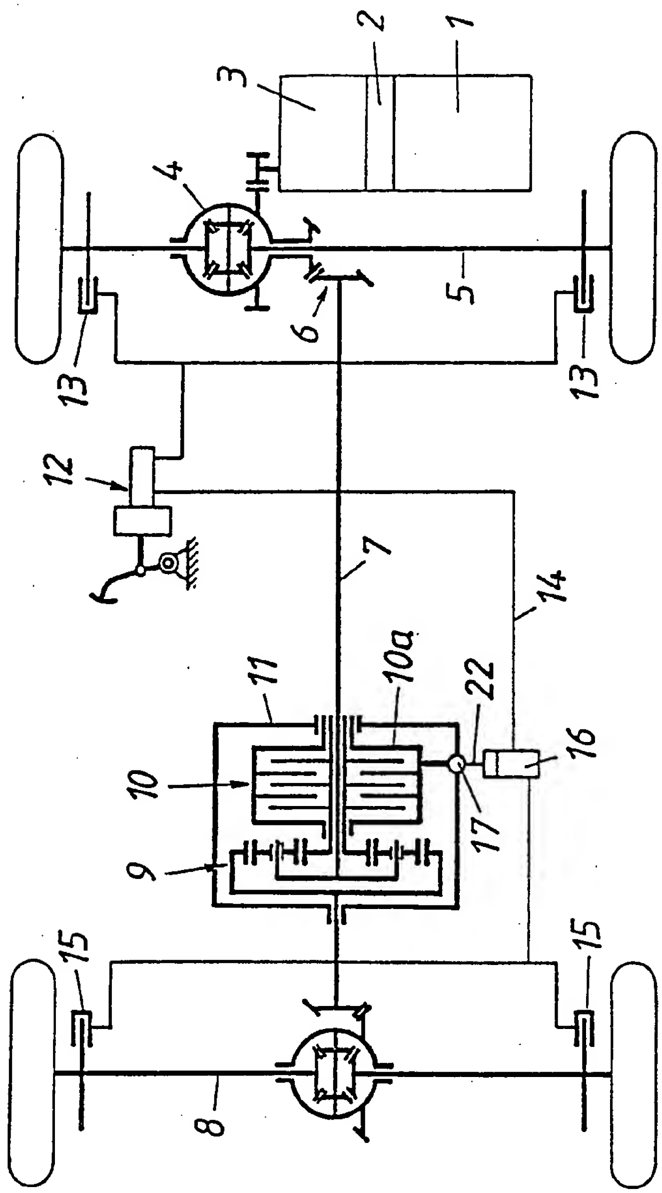


FIG. 1

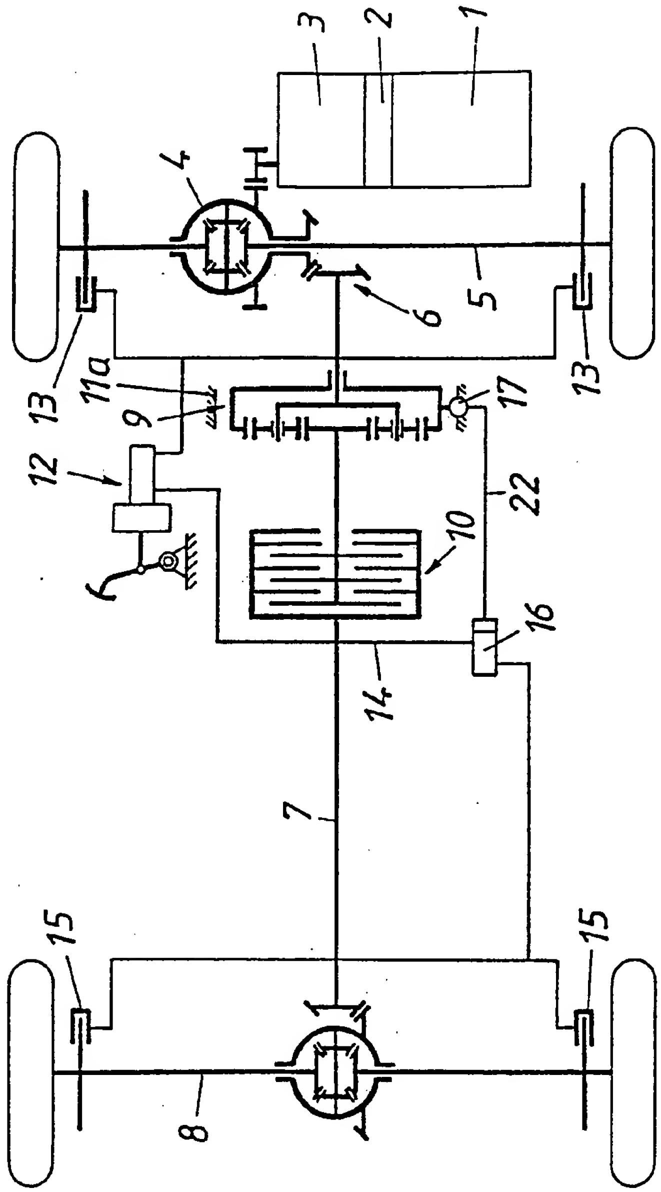


FIG. 2

FIG. 3

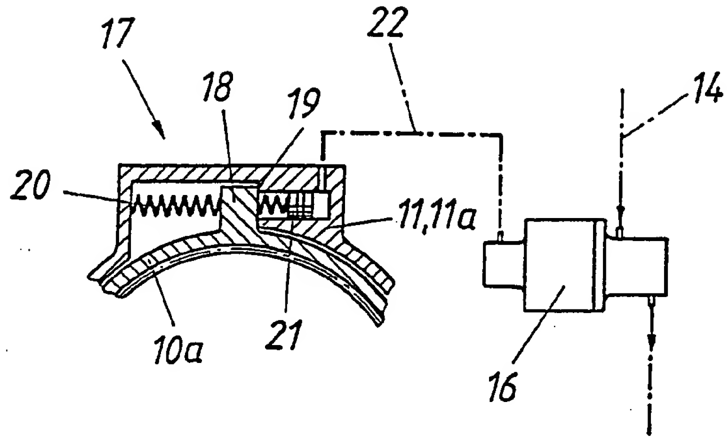
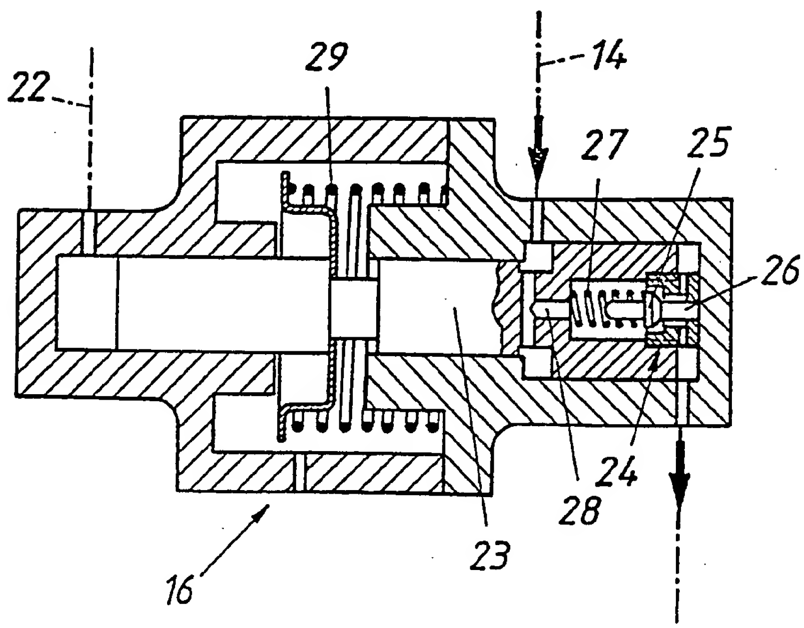


FIG. 4



MOTOR VEHICLE HAVING TWO DRIVEN AXLES AND BRAKE SYSTEM

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to a motor vehicle comprising two driven axles, and an interaxle power train, which operatively connects the two driven axles and comprises an interaxle coupling for transmitting torque only when the two coupling members of the coupling rotate at different speeds, and a speed-changing transmission in series with the coupling, and also comprising a brake system for braking all wheels of the driven axles.

2. Description of the Prior Art

In a motor vehicle, particularly in a passenger car, the brake system is usually so designed that the adhesion is utilized to a higher degree at the front axle than at the rear axle so that the wheels of the rear axle will not be blocked before the wheels of the front axle as a blocking of the wheels of the rear axle would eliminate the directional control of the vehicle. That utilization of the adhesion at the wheels of both axles to different degrees involves different wheel slips of the two axles. In a motor vehicle which has two driven axles, which are operatively connected by a hydraulic friction coupling or any interaxle coupling which transmits torque only when the coupling members of the coupling rotate at different speeds, the different wheel slips of the two axles will result in a differential speed so that torque will be transmitted by the interaxle coupling. Because a braking will cause the wheels of the driven front axle to rotate with a larger slip, i.e., at a lower speed, than those of the driven rear axle, the torque will be transmitted by the interaxle coupling from the driven rear axle to the driven front axle. Owing to that transmission of torque, the adhesion at the wheels of the driven rear axle will be utilized to a higher degree and the adhesion at the front wheels will be utilized to a lower degree so that said degrees of utilization will approach each other. For a given adhesion, a careful control of the brake pressure will permit a higher retardation of a vehicle having two driven axles than of a vehicle having only one driven axle because in the former the wheels of the rear axle will be braked more strongly and the transmission of torque from the rear axle to the front axle will delay the blocking of the wheels of the front axle. In case of a blocking of the wheels of the driven front axle, the high differential speed will cause the hydraulic friction coupling or other interaxle coupling between the two driven axle to transmit a higher torque so that the adhesion at the wheels of the rear axle will immediately be overcome, at least one of the wheels of the rear axle will be blocked and the directional control will be lost. That riding and braking behavior of the vehicle will not be altered when the interaxle power train comprises also a speed-changing transmission, e.g., a planetary gear train, in series with interaxle coupling.

From U.S. Pat. No. 4,605,087 it is already known to provide a drive system which has a hydraulic friction coupling connected between the two driven axles and in which in case of a blocking of the front wheels a blocking of the wheels of the rear axle will be prevented by a clutch or an overrunning coupling, which is connected between the two axles, so that a reduction of the speed of the front wheels resulting from an application of the service brake will not be transmitted by the power train to the rear wheels. In that arrangement the advantages afforded by the transmission of torque between the two driven axles in case of a careful application of the brake are eliminated.

SUMMARY OF THE INVENTION

It is an object of the invention to eliminate said disadvantages and to provide a motor vehicle which is of the kind described first hereinbefore and in which the advantages afforded by the transmission of torque between the two driven axles during a normal braking are preserved and the tendency of the wheels of the rear axle to block during a blocking of the front wheels is reduced.

This object is accomplished in accordance with the invention in that the brake pressure regulator, which is associated in known manner with the rear axle is arranged to be xontrolled by a pressure-setting controller, which is responsive to a reversal of the direction in which torque is transmitted to and from the rear driven axle and in case of a transmission of torque from the rear driven axle and in dependence on the torque being transmitted will cause the pressure applied to brake the wheels of the rear axle to be substantially reduced relative to the pressure applied to brake the wheels of the driven front axle.

In that arrangement a reversal of the direction in which torque is transmitted to and from the driven rear axle is utilized to adjust the brake pressure regulator in such a manner that the pressure applied to brake the wheels of the rear axle is lower than the pressure applied to brake the wheels of the front axle as soon as torque is being transmitted from the rear axle to the front axle and the torque transmitted by the interaxle coupling has risen to a predetermined value. The reduction of the pressure applied to brake the wheels of the rear axle will eliminate the risk of the blocking of said rear wheels in such a manner that the advantages afforded by the transmission of torque between the two driven axles will be preserved during normal operation because the pressure applied to brake the wheels of the rear axle will not be reduced until the torque transmitted by the interaxle coupling has risen to a predetermined value.

If the speed-changing transmission connected in series with the interaxle coupling consists of a planetary gear train hving a torque-transmitting element for transmitting a reaction torque, that torque-transmitting element or that coupling member of a liquid friction coupling to which said reaction torque is directly transmitted may be so arranged that during a transmission of torque to the driven rear axle said torque transmitting element or said coupling member of the interaxle coupling will be supported against a stop which is rigid with the chassis of the vehicle and in case of a reversal of the direction of torque transmission said torque-transmitting element or said coupling member of the interaxle coupling is disengaged from said stop against the force of a spring so that said torque-transmitting element or said coupling constitutes at least part of the pressure-setting controller for controlling the brake pressure regulator. In that arrangement, the advantage afforded by the direct measurement of the magnitude and direction of the torque being transmitted is utilized. Simple structural means may be utilized in that case because they are required to take up only the reaction torque. The operative connection to the brake pressure regulator may be established by mechanical or electri-

cally operated means and is preferably established by a fluid-handling system.

When the vehicle is strongly retarded from a high speed, torque will be transmitted from the driven rear axle to the driven front axle and this will result in a reduction of the pressure applied to brake the wheels of the rear axle. But such a strong retardation can occur in any case only when the adhesion is relatively high so that in that case a control resulting in a reduction of the pressure applied to brake the wheels of the driven rear axle is not desirable. For this reason it is within the scope of the invention to provide an auxiliary device for interrupting the operative connection between the pressure-setting controller and the brake pressure regulator in response to a rise of the retardation of the vehicle above a predetermined value.

A blocking of the wheels of the rear axle and the resulting loss of directional control usually result in a yaw of the vehicle, i.e., in an undesired lateral angular movement about the vertical axis of the vehicle, and/or in a swerving of the vehicle. Within the scope of the invention it will be sufficient to prevent a blocking of the wheels of the rear axle by a reduction of the pressure applied to brake said wheels when there is actually a risk of an uncontrolled yawing movement. For this reason it is within the scope of the invention to provide an operative connection between the pressure-setting controller and the brake pressure regulator only when the lateral angular acceleration of the vehicle has risen above a predetermined value.

Conventional brake pressure regulators associated with the wheels for braking the wheels of the driven rear axle are usually so designed that they will freely transmit the inlet pressure only until said pressure has risen to a critical value, which is predetermined or can be selected. When the inlet pressure has risen above said critical value, only a certain percentage of the rising input pressure is transmitted to the brakes for the wheels of the rear axle. In such a system, there will always by a pressure at the outlet of the brake pressure regulator when a pressure is applied to its inlet. In order to permit the pressure in the lines leading to the brakes for the wheels of the driven rear axle to be reduced substantially to the atmospheric pressure, the brake pressure regulator is provided with means for reducing to the pressure in the line leading to the brakes for the wheels of the rear axle substantially to the atmospheric pressure independently of the inlet pressure applied to the brake pressure regulator.

BRIEF DESCRIPTION OF THE DRAWING

FIG. 1 is a diagrammatic representation of the drive system of a motor vehicle.

FIG. 2 is a diagrammatic representation of a modified drive system.

FIG. 3 is a simplified view showing partly in section a detail.

FIG. 4 is an axial sectional view showing a brake pressure regulator on a larger scale.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The invention is illustrated by way of example on the drawing.

From a motor 1 of a vehicle, torque is transmitted via a driver-controlled clutch 2 and a change-speed gearbox 3 to the housing of a differential 4 for the front axle 5 and via a bevel gear train 6 to an interaxle power train 7 for driving the rear axle 8. In the embodiment shown in FIG. 1 the interaxle power train 7 includes a planetary gear train 9, which has a planetary gear train 9, which has a planet carrier that is directly driven by the bevel gear train 6 and an internal gear for driving the rear axle 8. The sun gear of that planetary gear train 9 is non-rotatably connected to one coupling member, specifically to the inner coupling member, of a liquid friction coupling 10. The other coupling member or housing 10*a* of that liquid friction coupling 10 is rotatably mounted in a gear housing 11, which is fixed to the chassis of the vehicle. From a main brake cylinder 12, pressure is applied directly to brakes 13 for the front wheels. A line 14 leading from the main brake cylinder 12 to the brakes 15 for the wheels of the rear axle 8 incorporates a brake pressure regulator 16, which is controlled by a pressuresetting controller 17.

The pressure-setting controller 17 comprises a lug 18 (FIG. 3), which is rigid with the housing 10*a* of the liquid friction coupling 10 and during a transmission of torque in the normal direction from the driven front axle 5 to the driven rear axle 8, i.e., when a torque in a clockwise sense in FIG. 3 is exerted on the housing 10*a*, said lug 18 bears on a fixed stop 19 that is rigid with the housing 11. A spring 20 is provided, which urges the lug 18 against the stop 19. In response to a reversal of the direction in which torque is transmitted and to a rise of the reaction torque to a value at which the force of the spring 20 is overcome, the lug 18 will disengage the stop 19. A piston 21 is subjected to hydraulic pressure from line 22 and in response to said disengagement of the lug 18 from the stop 19 is displaced to the left in FIG. 3 so that a pressure drop is effected in the line 22, which leads to the brake pressure regulator 16.

The brake pressure regulator 16 shown in FIG. 4 comprises a housing and a piston 23, which is displaceable in said housing and at one end is provided with a valve 24. The valve 24 comprises a valve disc 25, which is biased by a spring 27 and carries a stem 26, which extends through the adjacent end wall of the piston 23. At the beginning of a braking operation the piston 23 assumes the position shown in FIG. 4 so that pressure is freely applied from the main brake cylinder 12 through the line 14, the passages 28 in the piston 23 and the valve 24 to the brakes 15 for the wheels of the rear axle 8. The areas of the piston faces at opposite ends of the piston 23 and the force of a spring 29 which biases the piston 23 are so matched that a pressure rise in the line 14 to a predetermined value will cause the piston 23 to be displaced to the left in FIG. 4 so that the spring 27 can now move the valve disc 25 against the associated valve seat and the flow through the piston 23 is thus interrupted. From that time, only a certain percentage of the pressure applied by the main brake cylinder 12 is transmitted to the brakes 15 for the wheels of the rear axle 8. That percentage will depend on the equal and opposite forces acting on the piston 23.

When torque is being transmitted from the rear axle 8 to the front axle 5 and said torque has risen to a predetermined value, the piston 21 of the pressure-setting controller 17 will effect a pressure reduction in the line 22 and also at the left-hand end of the piston 23 in FIG. 4. As a result, the pressure in the line leading from the brake pressure regulator 16 to the brake 15 for the wheels of the rear axle 8 will also be reduced. If the piston 23 is integral and the spring 29 is properly dimensioned, the pressure in line 22 may be reduced to such an extent that the piston 23 is displaced to the left in FIG.

4 to such an extent that the pressure in the line leading to the brakes **15** for the wheels of the driven rear axle **8** is reduced substantially to the atmospheric pressure.

In the embodiment shown in FIG. **2** the internal gear of the planetary gear train **9** is supported by the pressure-setting controller **17** on a member **11***a*, which is rigid with the chassis, and the liquid friction coupling **10** serves as an interaxle differential between the driven front axle **5** and the driven rear axle **8** so that said coupling is not supported against rotation.

It may be desirable to interpose in the connection between the pressure-setting controller **17** and the brake pressure regulator **16** a sensor which utilizes, e.g., an inert mass for interrupting said connection when the retardation of the vehicle has risen above a predetermined value or when the lateral angular acceleration of the vehicle is below a predetermined value.

What is claimed is:

1. In a motor vehicle comprising
   a driven front axle having front wheels,
   a driven rear axle having rear wheels,
   an interaxle power train interconnecting said front and rear axles and comprising a speed-changing transmission and in series with said transmission a liquid friction coupling having two coupling members and adapted to transmit torque only when said two coupling members rotate at different speeds, and
   a brake system for applying pressure to brake said front and rear wheels, which brake system comprises a brake pressure regulator for controlling the pressure applied to brake said rear wheels,
   the improvement residing in that
   a pressure-setting controller is provided, which is responsive to a reversal of the direction of torque transmission between said interaxle coupling and said rear axle and to the value of torque transmitted from the rear axle to said interaxle coupling and is adapted to operate said brake pressure regulator in dependence on said value of torque said brake pressure regulator so as to effect a substantial reduction of the pressure applied to brake said rear wheels relative to the pressure applied to brake said front wheels only during a transmission of torque from said rear wheels to said interaxle coupling.

2. The improvement set forth in claim **1**, wherein said pressure-setting controller is adapted to effect said reduction of pressure only when said torque transmitted from said rear axle to said interaxle coupling has risen to a predetermined value.

3. The improvement set forth in claim **1**, wherein said pressure-setting controller is adapted to effect said reduction of pressure to an extent which depends on said value of torque.

4. The improvement set forth in claim **1**, as applied to a motor vehicle which comprises a chassis and in which said speed-changing transmission comprises a planetary gear train having a torque-transmitting member for directly transmitting a reaction torque to one of said coupling members, wherein
   one of the elements consisting of said torque-transmitting member and said one coupling member constitutes a part of said pressure-setting controller,
   said pressure-setting controller compriss a stop, which is rigid with said chassis,
   said one element is arranged to bear on said stop during a transmission of torwue from said interaxle coupling to said rear axle and is adapted to be disengaged from said stop during a transmission of torque from said rear axle to said interaxial coupling,
   spring means are provided, which urge said one element toward said stop, and
   said pressure-setting controller comprises means for operating said brake pressure regulator in response to the disengagement of said one element from said stop so as to effect a substantial reduction of the pressure applied to brake said rear wheels relative to the pressure applied to brake said front wheels.

5. The improvement set forth in claim **1**, wherein
   means are provided for establishing an operative connection between said pressure-setting controller and said brake pressure regulator and
   a retardation sensor is provided for sensing the retardation of said vehicle and for interrupting said operative connection when said retardation sensor senses a retardation above a predetermined value.

6. The improvement set forth in claim **1**, wherein
   means are provided for establishing an operative connection between said pressure-setting controller and said brake pressure regulator and
   a yaw sensor is provided for sensing the lateral angular acceleration of said vehicle and for interrupting said operative connection when said yaw sensor senses a lateral angular acceleration below a predetermined value.

7. The improvement set forth in claim **1** as applied to a vehicle in which said brake pressure regulator has an inlet and an outlet, wherein
   said brake pressure regulator comprises means for reducing under the control of said pressuresetting controller the pressure at said outlet substantially to the atmospheric pressure regardless of the pressure at said inlet.

* * * * *

United States Patent [19]

Iwata et al.

[11] Patent Number: 4,805,718

[45] Date of Patent: Feb. 21, 1989

[54] POWER TRANSMISSION SYSTEM FOR A FOUR-WHEEL DRIVE

[75] Inventors: Seiichi Iwata; Kyoji Takenaka, both of Tokyo, Japan

[73] Assignee: Fuji Jukogyo Kabushiki Kaisha, Tokyo, Japan

[21] Appl. No.: 892,523

[22] Filed: Aug. 1, 1986

[30] Foreign Application Priority Data

Aug. 5, 1985 [JP] Japan 60-172138

[51] Int. Cl.[4] B60K 17/34

[52] U.S. Cl. 180/233; 180/247; 192/46; 192/48.92; 192/51

[58] Field of Search 180/233, 247, 249, 250, 180/23, 236, 248; 192/51, 50, 48.92, 48.1, 46, 43.1, 43, 47; 364/424.1

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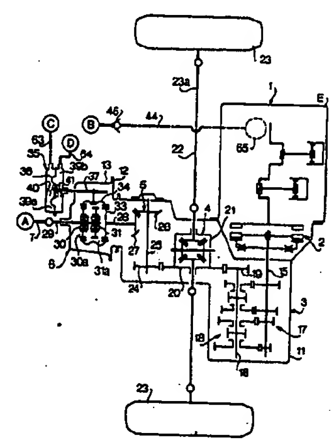
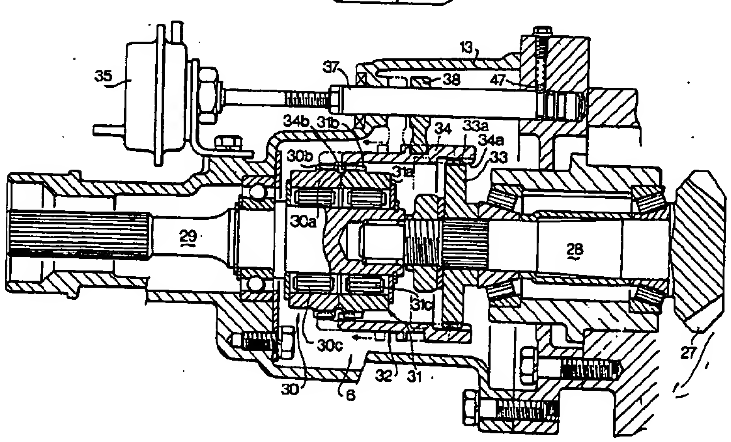
Primary Examiner—David M. Mitchell
Assistant Examiner—Mitchell J. Hill
Attorney, Agent, or Firm—Martin A. Farber

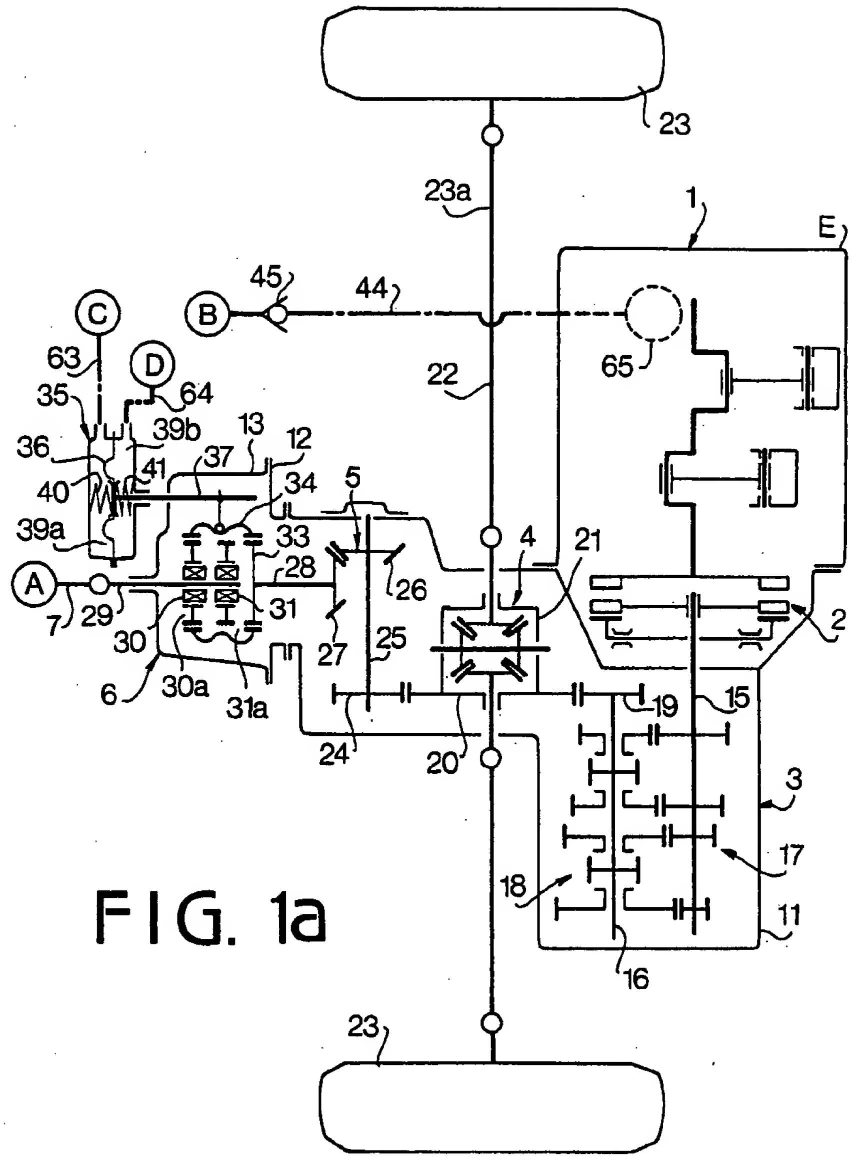
[57] ABSTRACT

A power transmission system for a four-wheel drive vehicle has front and rear wheel driving power transmission systems for transmitting the output of a transmission. A forward overrunning clutch and a reverse overrunning clutch are provided in one of the power transmission systems. One of the overrunning clutches can be selected in accordance with the operation of a driver of the vehicle. The overrunning clutch operates to transmit the power to corresponding wheels and to allow the selected wheels to rotate faster than the other wheels. A locking device is provided for locking both overrunning clutches.

11 Claims, 4 Drawing Sheets



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23a
1
E
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64
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39b
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12
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34
5
40
39a
33
4
21
28
A
26
7
29
31
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30
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25
6
30a
31a
19
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3
17
18
11
16
23

FIG. 1a

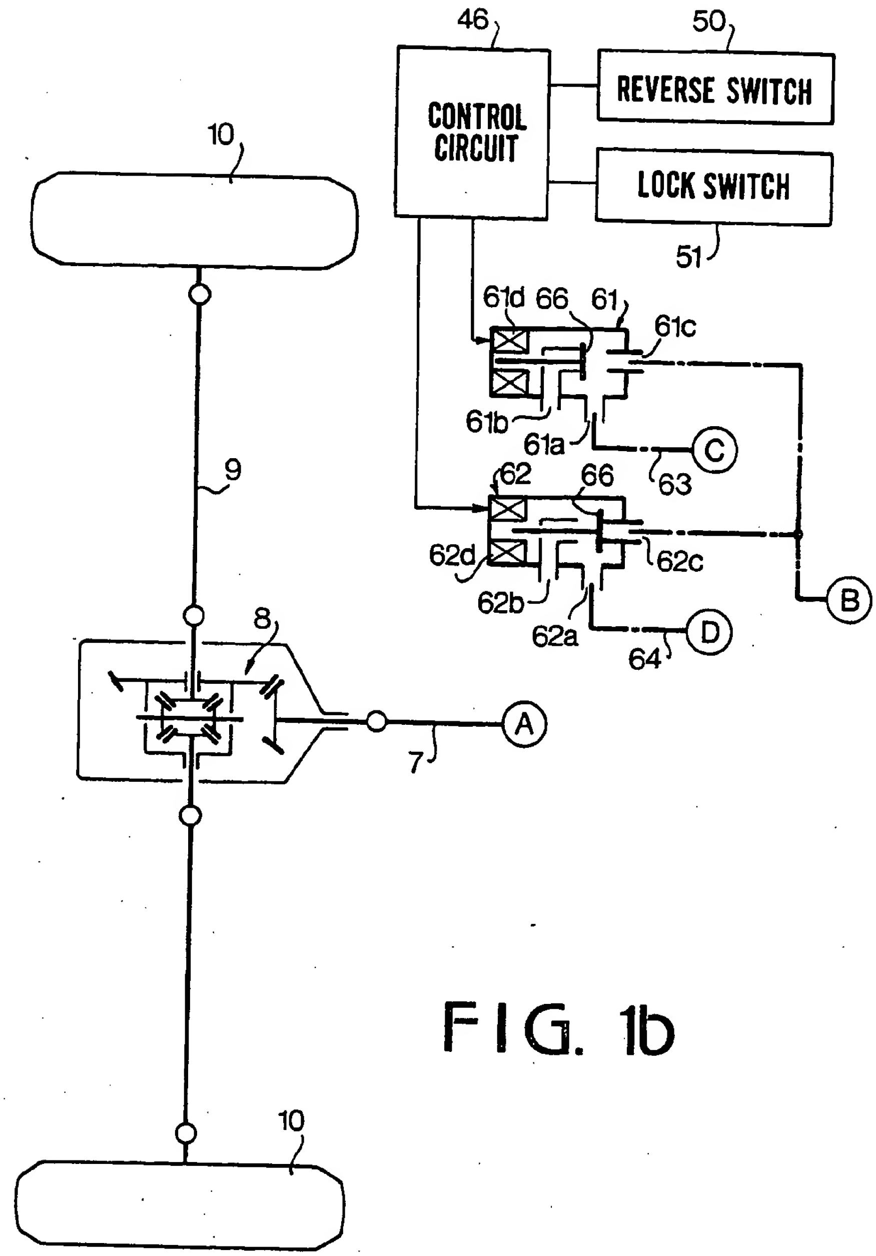
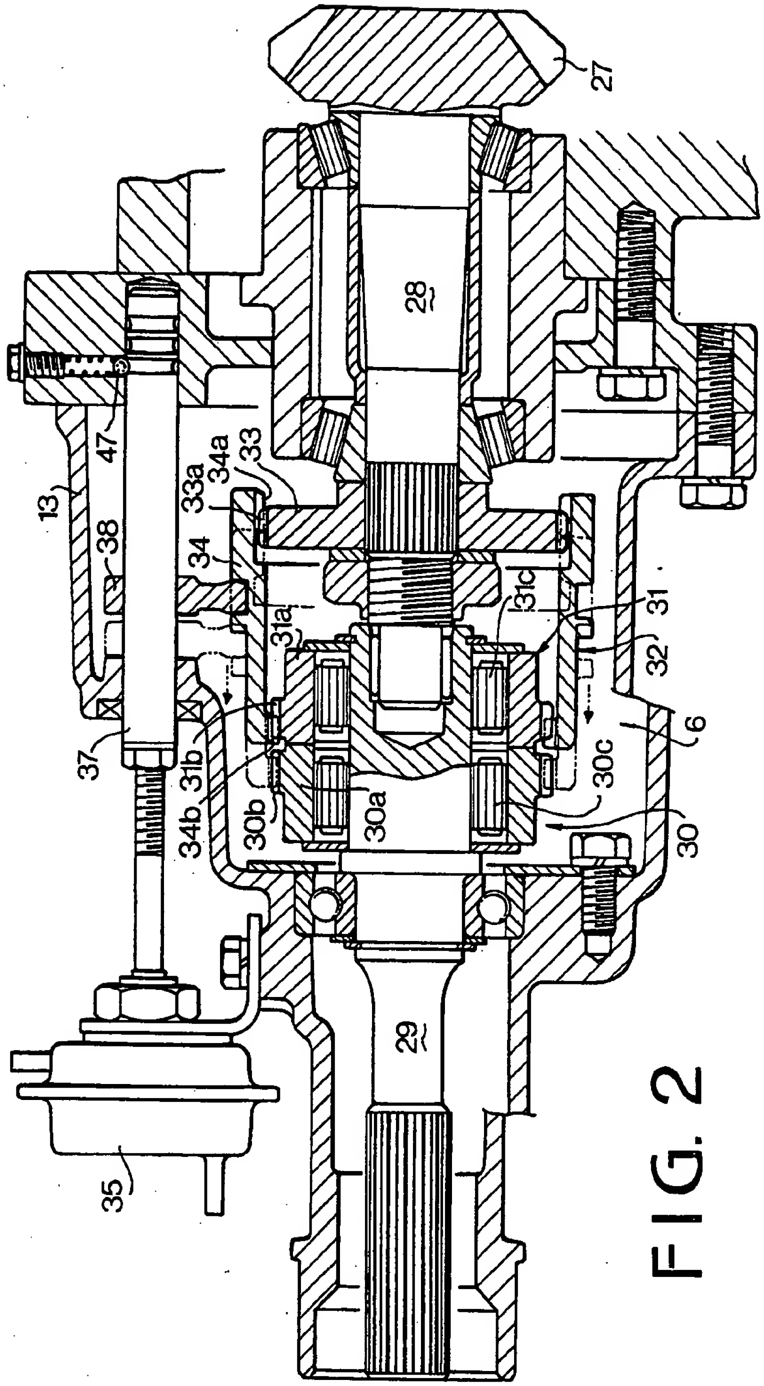

46
50
CONTROL CIRCUIT
REVERSE SWITCH
LOCK SWITCH
51
10
61d 66 61
61c
61b
61a
C
63
9
62
66
62d
62c
B
62b
62a
D
64
8
A
7
10

FIG. 1b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28
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47
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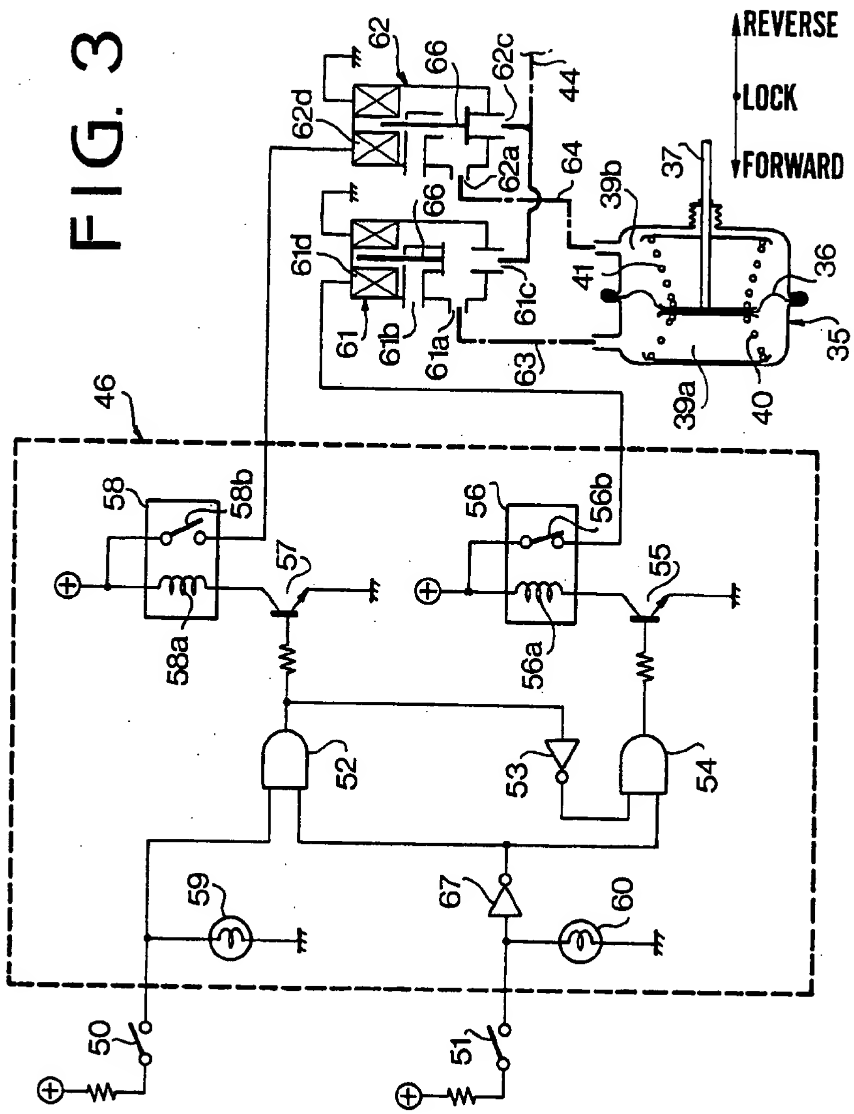
33a
34a
33
34
31a
31c
31
32
6
37
31b
34b
30b
30a
30c
30
29
35

FIG. 2

FIG. 3



04/05/2004, EAST Version: 1.4.1

POWER TRANSMISSION SYSTEM FOR A FOUR-WHEEL DRIVE

BACKGROUND OF THE INVENTION

The present invention relates to a power transmission system for a full-time four-wheel drive vehicle having a mechanism for absorbing the difference between the speeds of the front and rear wheels.

In a part-time four-wheel drive vehicle, the power transmission system can be selectively converted from the two-wheel driving mode to the four-wheel driving mode by engaging a clutch which is manually operated by a select lever.

The two-wheel drive is selected on paved dry surfaces. The reason is as follows. When the vehicle negotiates corners, the front wheels run through an arc of greater radius than that of the rear wheels and therefore tend to rotate faster than the rear wheels.Such a difference between the speeds the of the front and rear wheels causes the braking of the vehicle, known as "tight corner braking". In order to prevent such a braking phenomenon, a full-time four-wheel drive vehicle, a transmission system of which is automatically changed to a two-wheel drive transmission system at a large steering angle, is disclosed in Japanese patent Laid Open No. 57-15019.

Further, Japanese Utility Model Laid Open No. 58-136829 discloses a full-time four-wheel drive tractor having a transmission system in which a one-way clutch (overrunning clutch) is provided in the power transmission for the front wheels. In a normal driving state, the one-way clutch does not engage, thereby establishing the two-wheel driving mode by the rear wheels. When the rear wheels skid, the one-way clutch engages to provide the four-wheel driving mode. However, the system can not prevent the tight corner braking at sharp corners, since the front wheels run faster than the rear wheels, thereby engaging the clutch.

SUMMARY OF THE INVENTION

The object of the present invention is to provide a power transmission system for a full-time four-wheel drive vehicle, which prevents the tight corner braking at corners by providing a double overrunning clutch system, and to provide a system for locking the double overrunning clutch so as to prevent skidding of the wheels.

According to the present invention, there is provided a power transmission system for a four-wheel drive vehicle having a transmission, comprising, first and second power transmission systems for transmitting the output of the transmission to front and rear wheels respectively, a forward overrunning clutch and a reverse overrunning clutch provided in one of the power transmission systems and effective for forward drive and reverse drive respectively, a selecting device for selecting one of the clutches in order to transmit the output of the transmission to corresponding wheels through the selected overrunning clutches. A lock device is provided for locking the forward and reverse overrunning clutches.

In an aspect of the invention, each of the forward and reverse overrunning clutches comprises an outer race operatively connected to a drive shaft of the transmission and rollers disposed between the outer race and a driven shaft.

The other objects and features of this invention will become understood from the following description with reference to the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIGS. *1a* and *1b* show a schematic illustration showing a power transmission system according to the present invention;

FIG. 2 is a sectional view of a main portion of the system; and

FIG. 3 is a circuit for operating an actuator for a double overrunning clutch.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring to FIGS. *1a* and *1b*, an engine unit E is transversely mounted on a rear portion of a vehicle. The engine unit has an engine 1, clutch 2 and transmission 3. The output of the transmission 3 is transmitted to rear wheels 23 of the vehicle through a differential 4 and axles *23a*. The output of the transmission 3 is also transmitted to front wheels 10 of the vehicle through a front wheel driving power transmission system comprising a transfer device 5, double overrunning clutch 6, propeller shaft 7, differential 8, and axles 9. A case 13 of the clutch 6 is secured to a case 11 of the transmission 3 through an intermediate case 12. The transmission 3 comprises an input shaft 15, output shaft 16, a plurality of change-speed gears 17, and synchronizers 18. An output gear 19 engages with a ring gear 20 secured to a case 21 of the differential 4. The transfer device 5 comprises a gear 24 engaged with ring gear 20, bevel gear 26 on a shaft 25 of gear 24, and bevel gear 27 engaged with bevel gear 26.

Referring to FIG. 2, a transfer drive shaft 28 of the bevel gear 27 and a driven shaft 29 connected to the propeller shaft 7 are co-axially disposed, and both shafts 28 and 29 are coupled through the double overrunning clutch 6. The double overrunning clutch 6 comprises a forward overrunning clutch 30 and a reverse overrunning clutch 31 each of which is in the form of a free-wheel. Clutches 30 and 31, comprise outer races *30a*, *31a* and rollers *30c*, *31c* disposed between the outer races *30a*, *31b* and the shaft 29, respectively. A forward-/reverse selecting device 32 is provided comprising an axially slidable sleeve 34 with toothing. A gear 33 is secured to the shaft 28. The sleeve 34 has two inside toothings *34a* and *34b*. The toothing *34a* is permanently engaged with a toothing *33a* of the gear 33. Depending on the axial position of the sleeve 34. The toothing *34b* is selectively engaged with a toothing *30b* of the forward clutch 30 or with a toothing *31b* of the reverse clutch 31, or simultaneously with both toothings *30b* and *31b*, the toothing *34b* having such a width that it can engage with both toothings *30b* and *31b* at the same time. Such a selection of these different toothing engagements is performed by shifting the sleeve 34 to one of three axial positions, namely, a forward position, a reverse position and a lock position. The forward overrunning clutch 30 is so arranged as to transmit the rotation of the outer race *30a* in the forward driving direction to the shaft 29 and as to allow the rotation of the shaft 29, in the forward driving direction, at higher speed than the outer race 30. The reverse overrunning clutch 31 is arranged oppositely. Namely, in the reverse driving direction, the rotation of the outer race *31a* is transmitted to the shaft 29, and in the reverse driving

direction to allow rotation of the shaft **29** at higher speed than the outer race **31b**.

The sleeve **34** has an annular groove on the periphery thereof, in which a shifter fork **38** is relatively rotatably slidably engaged. The shifter fork **38** is operatively connected to a diaphragm **36** (FIG. 1) in a vacuum operated actuator **35** through a rod **37**. A ball lock device **47** is provided to position the rod **37** at one of the three positions.

As shown in FIGS. 1a and 3, the diaphragm **36** of the actuator **35** is held by springs **40** and **41** on both sides thereof to an intermediate lock position when both vacuum chambers **39a** and **39b** defined by the diaphragm are communicated with the atmosphere as described hereinafter. The vacuum chambers **39a** and **39b** are communciated with ports **61a** and **62a** of solenoid operated valves **61** and **62**, respectively. The solenoid operated valves **61** and **62** have atmosphere ports **61b** and **62b**, vacuum ports **61c** and **62c**, and solenoids **61d** and **62d** respectively. Both vacuum ports **61c** and **62c** are communicated with an intake manifold **65** through a passage **44** having a check valve **45** so as to be supplied with vacuum pressure in the intake manifold **65**. Each valves has a valves body **66** for selectively closing the atmosphere port or vacuum port in accordance with energization of the solenoid. When the solenoid is energized by the output of a control circuit **46**, the valve body **66** closes the atmosphere port and opens the vacuum port to communicate the vacuum chamber of the actuator **35** with the intake manifold **65**.

Referring to FIG. 3 showing the control circuit **46**, there is provided a reverse switch **50** which is closed when a reverse gear in the transmission **3** is selected, and a lock switch **51** for locking the overrunning clutches **30** and **31**. The reverse switch is connected to an AND gate **52** and the lock switch **51** is connected to AND gates **52** and **54** through an inverter **67**. The output of the AND gate **52** is connected to a base of a transistor **57** and to the AND gate **54** through an inverter **53**. The output of the AND gate **54** is connected to a base of a transistor **55**. The transistor **57** is connected between a coil **58a** of a relay **58** and the ground, and the transistor **55** is connected between a coil **56a** of a relay **56** and the ground. A contact **58b** of the relay **58** and a contact **56b** of the relay **56** are connected to solenoids **62d** and **61d**, respectively.

In the state shown in FIG. 3, the output of the AND gate **52** is at a low level and the output of the AND gate **54** is at a high level, thereby turning on the transistor **55**. Accordingly, the coil **56a** of the relay **56** is excited to turn on the contact **56b**, so that the solenoid **61d** of the valve **61** is energized to open the vacuum port **61c**. The vacuum chamber **39a** of actuator **35** is communicated with the intake manifold **65** through passage **63**, valve **61** and passage **44**, and the diaphragm **36** is deflected to the left by the vacuum pressure in the intake manifold to draw the rod **37** to the left (forward position). Thus, the sleeve **34** is shifted to the left, so that the toothing **34b** engages with the toothing **30b** of the forward overrunning clutch **30**. Accordingly, when the clutch **2** is engaged, the output of the change-speed transmission **3** is transmitted to the front wheels **10** through shaft **29**, forward overrunning clutch **30**, shaft **29** and propeller shaft **7**. When the vehicle turns a corner, the front wheels **10** rotate faster than the rear wheels **23**. That is, the shaft **29** rotates faster than the outer race **30a** (in advance of it). However, the overrunning clutch **30** permits such a faster rotation of the shaft **29**. Thus, the difference between the speeds of the front and rear wheels is absorbed in the clutch, and the vehicle turns the corner smoothly without the braking phenomenon.

When the transmission **3** is changed to the reverse driving state, the reverse switch **50** is closed. Accordingly, the output of AND gate **52** goes to a high level, the output of AND gate **54** goes to a low level, and a lamp **59** is turned on. Thus, the transistor **57** is turned on to close the contact **58b** to energize the solenoid **62d**. Accordingly, the vacuum chamber **39b** is communicated with the intake manifold **65**, shifting the rod **37** to the right (reverse position). Thus the reverse overrunning clutch **31** is selected in the same manner as described above but for the reverse driving. In reverse driving, the same operation as the forward driving is performed.

When the lock switch **51** is closed, a lamp **60** is lighted up and the outputs of AND gates **52** and **54** become low. Thus, transistors the **55**, **57** become nonconductive, thereby de-energizing the solenoids **61d**, **62d** to communicate the vacuum chambers **39a**, **39b** with the atmosphere. The diaphragm **36** is located at the neutral position by springs **40**, **41** to position the rod **37** at the lock position. Accordingly, the toothing **34b** of the sleeve **34** engages both toothings **30b** and **31b** to lock the double overrunning clutch C. Thus, a direct connected four-wheel driving mode is established. Therefore, the vehicle is safely driven by four wheels without skidding of the wheels.

While the presently preferred embodiment of the present invention has been shown and described, it is to be understood that this disclosure is for the purpose of illustration and that various changes and modifications may be made without departing from the spirit and scope of the invention as set forth in the appended claims.

What is claimed is:

1. A power transmission system for a four-wheel drive vehicle having front and rear wheels and a transmission, the system comprising:
   - a power transmitting system for transmitting the output of the transmission to the front and rear wheels;
   - a forward overrunning clutch and a reverse overrunning clutch provided in said power transmitting system and when selected effective for forward drive and reverse drive respectively;
   - means for selecting alternate ones of said overrunning clutches at a time for transmitting the output of the transmission through the selected one of said overrunning clutches, said selected one of said overrunning clutches at the time being arranged so as to permit overrunning of said selected overrunning clutch; and
   - said means further for selecting and locking together both said forward and reverse overrunning clutches for transmitting the output of the transmission through one of said clutches and such that the other of said clutches makes overrunning of said one of said clutches impossible.

2. The power transmission system according to claim 1, wherein
   - each of the forward and reverse overrunning clutches comprises an outer race selectively operatively connected via said means to a drive shaft of said power transmitting system, the drive shaft being connected from the transmission, and rollers operatively connected to the outer race and the rollers of both clutches being connected to a same driven

shaft of said power transmitting system, the driven shaft being connected to said front wheels, said drive and driven shafts being in said power transmitting system between said front and rear wheels.

3. The power transmission system according to claim 2, wherein

said means comprises a sleeve connected to the drive shaft and axially displaceable into three respective positions for selecting alternate ones of said overrunning clutches at a time and for selecting and locking together both said clutches, respectively by connection of said sleeve to corresponding of said outer races, and a vacuum operated actuator having a diaphragm, a rod connected to the diaphragm and the sleeve, solenoid operated valves for selectively supplying vacuum pressure to the actuator, and an electric circuit for operating said solenoid operated valves in dependency on a selected position so as to shift the rod corresponding to one of the three positions.

4. The power transmission system according to claim 1, wherein

said power transmitting system includes a portion between the front wheels and the rear wheels of the vehicle, said portion including a driven shaft connected to one of the front wheels and the rear wheels of the vehicle and a drive shaft connected to the transmission.

each said overrunning clutch having a first member and a second member, said first member being operatively connected to said second member so as to be rotated by said second member in a first direction, and rotatable in said first direction faster than the speed at which said second member rotates said first member, said first direction of said first and second clutches being opposite rotatable directions,

said first members being connected to said one of said shafts, and

said means is for selectively engaging said second members with the other of said shafts, together and individually respectively, said clutches being arranged such that when said second members are engaged together with said other shaft, said first member of respective of the clutches is prevented from being rotatable in said first direction faster than the speed at which said second member rotates said first member.

5. The power transmitting system according to claim 1, further comprising

first switch means operatively connected to said means for automatically selecting said forward and reverse overrunning clutches upon occurrence of forward and reverse transmission ranges, respectively, of the transmission, and

second switch means operatively connected to said means for initiating the selecting and locking of both said overrunning clutches.

6. A power transmission system for a four-wheel drive vehicle having a change-speed transmission having a forward transmission range and a reverse transmission range, the system further having first and second power transmission systems for transmitting output of the change-speed transmission respectively to front and rear wheels of the vehicle, and a drive shaft and a driven shaft in one of said first and second power transmission systems, and the drive shaft being connected to the output of the transmission, the system comprising;

first means comprising a forward overrunning clutch and second means comprising a reverse overrunning clutch selectively coupling said drive and driven shafts,

the first means comprising said forward overrunning clutch for permitting rotation of the front wheels faster than the rear wheels during forward driving in said forward transmission range when said forward overrunning clutch alone is selected for coupling said drive shaft to said driven shaft,

each of the forward and reverse overrunning clutches comprising an outer race and rollers couplingly disposed between the outer race and the driven shaft,

first toothing provided on a periphery of the drive shaft,

selector means for selecting said clutches for coupling said drive and driven shafts, said selector means comprising an axially displaceable sleeve having an inside second toothing rotatably engaged with the first toothing in axially displaced positions of the sleeve,

the sleeve having an inside third toothing,

fourth and fifth toothings provided on peripheries of the outer races of the forward and reverse overrunning clutches, respectively,

the third toothing being so arranged as to selectively rotatably engage individually with one of the fourth and fifth toothings, respectively, and to rotatably engage simultaneously with both the fourth and fifth toothings, respectively in three respective axial positions of the sleeve, and

the second means comprising said reverse overrunning clutch for permitting rotation of the front wheels faster than the rear wheels during reverse driving in said reverse transmission range when said reverse overrunning clutch alone is selected for coupling said drive shaft to said driven shaft.

7. The power transmission system according to claim 6, wherein

the selector means further comprises shifting means for shifting the sleeve to a forward axial position and a reverse axial position in dependency on the selection for said coupling of the forward overrunning clutch and the reverse overrunning clutch, respectively, and for shifting the sleeve to an intermediate axial position for the coupling of both of said clutches simultaneously.

8. The power transmission system according to claim 7 wherein the shifting means comprises a vacuum operated actuator having a diaphragm, a rod connected to the diaphragm and the sleeve, solenoid operated valves for selectively supplying vacuum pressure to the actuator, and an electric circuit for operating said solenoid operated valves in dependency on a selected position so as to shift the rod corresponding to one of the three positions.

9. A power transmission system for a four-wheel drive vehicle having front and rear wheels and a transmission, the system comprising:

a power transmisting system for transmitting the output of the transmission to the front and rear wheels;

means, comprising a forward overrunning clutch and a reverse overrunning clutch each clutch being couplable into the power transmitting system between the front and rear wheels and arranged, and when either one of said clutches alone is individually coupled into the power transmitting system,

for transmitting the output of the transmission through the coupled of said forward overrunning clutch and said reverse overrunning clutch, respectively, in a forward driving direction and a reverse driving direction, respectively, and permitting the front wheels to turn faster than the rear wheels by overrunning of the coupled clutch;

coupling means for individually coupling respective of said clutches alone into the power transmitting system, and for coupling both of said clutches simultaneously into the power transmitting system, respectively;

said clutches are arranged such that when both clutches are simultaneously coupled into the power transmitting system they prevent the front wheels from running faster than the rear wheels and transmit the output of the transmission in the forward driving direction and reverse driving direction respectively.

10. The power transmitting system according to claim 9, further comprising

first switch means operatively connected to said coupling means for individually coupling said forward and reverse overrunning clutches upon occurrence of forward and reverse transmission ranges, respectively, of the transmission, and

second switch means operatively connected to said coupling means for initiating the simultaneous coupling of both said overrunning clutches.

11. The power transmitting system according to claim 9, wherein

said power transmitting system including a driven shaft connected to one of the front wheels and the rear wheels of the vehicle and a drive shaft connected to the transmission,

each said overrunning clutch having a first member and a second member, said first member being operatively connected to said second member so as to be rotated by said second member in a first direction, said rotatable in said first direction faster than the speed at which said second member rotates said first member, said first direction of said first and second clutches being opposite rotatable directions,

said first members being connected to said one of said shafts, and

said coupling means is for selectively engaging said second members with the other of said shafts, together and individually respectively, said clutches being arranged such that when said second members are engaged together with said other shaft, said first member of respective of the clutches is prevented from being rotatable in said first direction faster than the speed at which said second member rotates said first member.

* * * * *

United States Patent [19]

Sommer

[11] **Patent Number: 4,830,136**

[45] **Date of Patent: May 16, 1989**

[54] **FOUR-WHEEL DRIVE MOTOR VEHICLE**

[75] Inventor: **Hans D. Sommer**, Graz, Austria

[73] Assignee: **Steyr-Daimler-Puch Aktiengesellschaft**, Vienna, Austria

[21] Appl. No.: **68,952**

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[30] **Foreign Application Priority Data**

Jul. 1, 1986 [AT] Austria 1771/86

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[52] **U.S. Cl.** **180/233**; 180/197; 180/247

[58] **Field of Search** .... 180/197, 243, 247, 248, 180/249; 303/95, 98, 100, 102, 103; 340/52 R; 361/238, 242; 192/13

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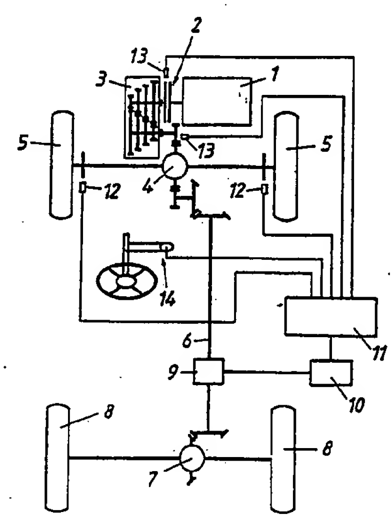
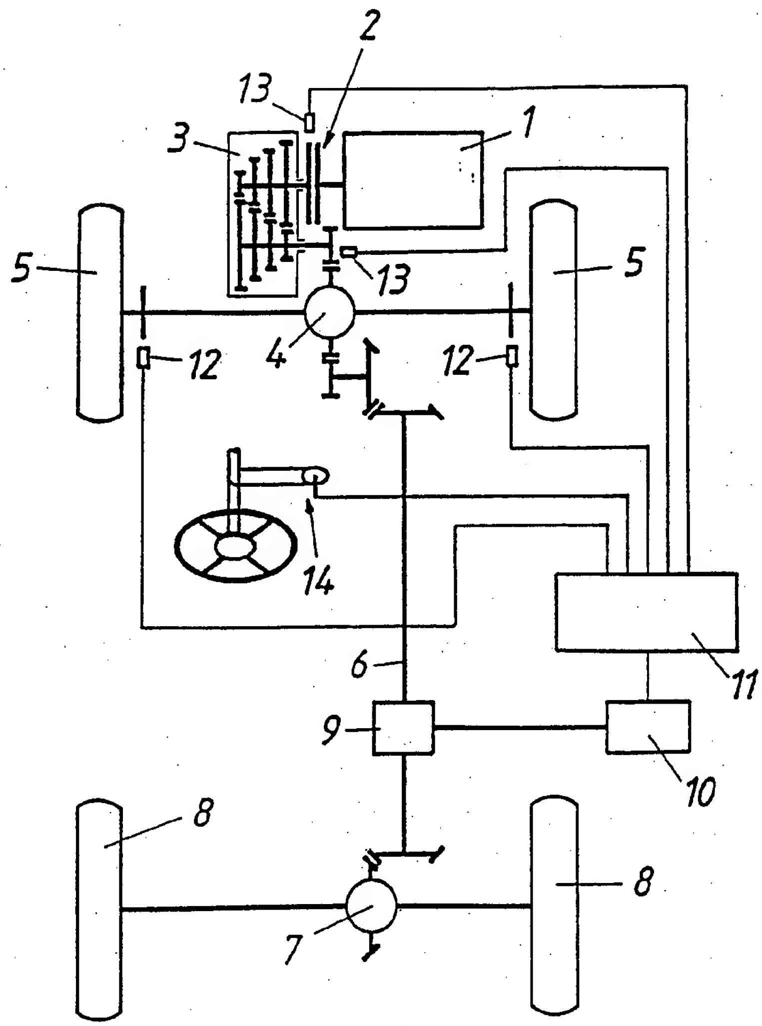
Primary Examiner—David M. Mitchell
Attorney, Agent, or Firm—Marmorek, Guttman & Rubenstein

[57] **ABSTRACT**

A four-wheel drive of a motor vehicle is arranged to effect automatic changes between a two-wheel drive mode and a four-wheel drive mode in dependence on the slip of two wheels (5) which are permanently driven. A frequent change from the four-wheel drive mode to the two-wheel drive mode for a speed comparison is to be avoided in a four-wheel drive which has a relatively simple structure. Two sensors (12) are associated with the two permanently driven wheels (5) and the output signals of said sensors are delivered to a computer (11), which determines for consecutive units of time the occurrences of the speed difference between said two wheels in excess of a predetermined threshold value and generates a distribution curve of said occurrences in consecutive units of time. That distribution curve is compared in the computer with a corresponding stored limiting curve, which is applicable to a minimum coefficient of road friction. The computer (11) controls a final control element (10) for effecting a change to the four-wheel drive mode when said distribution curve exceeds said limiting curve.

19 Claims, 1 Drawing Sheet

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FOUR-WHEEL DRIVE MOTOR VEHICLE

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to a four-wheel drive motor vehicle comprising means for effecting an automatic change from two-wheel drive mode to a four-wheel drive mode in dependence on the slip of the two permanently driven wheels.

2. Description of the Prior Art

It is already known to measure the slip of the two permanently driven wheels of a four-wheel driven motor vehicle in that the speeds of said two permanently driven wheels and of the optionally drivable wheels are compared and the four-wheel drive modes is selected in response to a high slip of the permenently driven wheels (FISITA Belgrade, June 2 to 6, 1986, Proceedings, Volume 2, pages 277 to 283). When the four-wheel drive mode has been selected, the optionally drivable wheels must be mechanically disconnected from the power transmission after a short interval of time so that the speeds of the permanently driven wheels and of the optionally drivable wheels can again be compared in order to ascertain whether or not the slip of the permanently driven wheels is still so large that a four-wheel drive is justified. Unless that mechanical disconnection is performed, there will be no speed difference between the permanently driven wheels and the optionally drivable wheels unless the power train leading to the optionally drivable wheels incorporate a differential or a liquid friction coupling. The clutch for connecting and disconnecting the optionally drivable wheels must meet special requirements and a differential will increase the structural expenditure involved and is not justified in some applications.

SUMMARY OF THE INVENTION

It is an object of the invention to eliminate the disadvantages stated above and so to improve a four-wheel drive motor vehicle which is of the kind described first hereinbefore that a need for a frequent change between a four-wheel drive mode and a two-wheel drive mode for measuring purposes and for a comparison of speeds will no longer be required, this is to be accomplished with structurally simple means.

That object is accomplished in accordance with the invention in that two sensors are respectively associated with the two permanently driven wheels and are used to detect the speed differences between said wheels and to deliver signals corresponding said speed differences to a computer, in which the occurrences of said speed differences in excess of a predetermined threshold value are derived from said signals, a distribution curve of said occurrences in consecutive units of time is generated and is compared with a limiting curve which is stored in said computer and applicable to a minimum coefficient of road friction, and a final control element is provided for selecting a four-wheel drive mode when the comparison indicates that the distribution curve exceeds the limiting curve.

The invention is based on the recognition that speed differences between the two permanently driven wheels will occur during any travel of the vehicle and that such speed differences will occur at a much lower frequency on a dry road having a high coefficient of road friction than, e.g., on a wet road having a low coefficient of friction resulting in a larger slip. The limiting curve stored in the computer is an empirically determined distribution curve for travel conditions which involve a limiting slip which just permits the vehicle to be driven in the two-wheel mode. But when the speed differences occur at a higher frequency, which is higher than that which is predetermined by the limiting curve which has been stored in the computer, this will indicate a high degree of slip is occurring at the two permanently driven wheels so that the four-wheel drive mode must be selected. As it is no longer necessary to compare the speed of the permanently driven wheels, on the one hand, and the speed of the optionally driven wheels, on the other hand, the four-wheel drive mode need not be interrupted frequently for a measurement. If the frequency of such excessive speed differences decreases below the values which are indicated by the limiting curve, the two-wheel drive mode will be restored because a decrease of said frequency will indicate a lower slip.

It will be understood that the travel conditions in the four-wheel drive mode will differ from those in the two-wheel drive mode and that this should be taken into account in the limiting curve which is used. For that purpose a second limiting curve is stored in the computer and is used for the change from the four-wheel drive mode to the two-wheel drive mode. That second limiting curve represents those values which must exceed the frequency of the detected excessive speed differences when a change from the four-wheel drive mode to the two-wheel drive mode is to be effected.

The motor vehicle may be so designed that the stored limiting curve or curves do not represent optimum values for an operation of the motor vehicle at all gear stages of the shiftable transmission of the motor vehicle. For this reason it may be desirable to store two limiting curves for each gear stage of the shiftable transmission in the computer and to select said limiting curves, for instance, in response to a comparison of the engine speed and the output speed of the transmission.

The two wheels of an axle will obviously rotate at different speeds when the vehicle is cornering and the resulting speed differences may not be due to different coefficients of friction. Particularly during a travel around long road bends that fact can be taken into account in that a sensor for detecting the steering angle is connected to the computer and the limiting curves are arranged to be varied in dependence on the steering angle detected by said sensor. Alternatively the final control element may maintain its state regardless of the state of the computer when the steering angle exceeds a predetermined upper limit.

BRIEF DESCRIPTION OF THE DRAWING

The drawing is a diagrammatic representation of an illustrative embodiment of a four-wheel drive of a motor vehicle in accordance with the invention.

The two front wheels **5** of a motor vehicle are permanently driven by an engine **1** through the intermediary of a clutch **2**, a speed-changing transmission **3** and a differential **4**. A power train **6** leads from the differential **4** to the interaxle differential **7** for driving the rear wheels **8**. The power train **6** incorporates an interaxle clutch **9** for effecting a change between the two-wheel drive mode and the four-wheel drive mode under the control of a clutch actuator **10**, which constitutes a final control element and is controlled by a computer **11**, which receives various output signals generated by

sensors. Two sensors **12** are associated with the two front wheels **5**, respectively, and detect the speeds of said front wheels **5** and deliver corresponding signals to the computer **11**, in which the occurrences of a difference between said two speeds in excess of a predetermined threshold value in consecutive units of time are detected and a corresponding distribution curve is generated. The gear stage which has been selected in the shiftable transmission **3** is detected by two sensors **13** for detecting the engine speed and the output speed of the transmission **3**, and by a comparator, which is incorporated in the computer **11** and compares the speeds detected by the sensors **13**. A sensor **14** for detecting the steering angle is also connected to the computer.

The computer may be designed to generate the above-mentioned distribution curve in dependence on the numbers of the occurrences of said excessive speed differences in consecutive units of time or in dependence on the numbers of consecutive units of time in which such excessive speed differences are detected.

I claim:

1. A motor vehicle comprising
first and second axles,
first and second permanently driven wheels mounted on said first axle,
third and fourth optionally driven wheels mounted on said second axle,
a four-wheel drive mode for selectively driving only said first and second wheels in a two-wheel drive mode and said first, second, third and fourth wheels in a four-wheel drive mode,
means for detecting intra-axle speed differences between said first and second permanently driven wheels on said first axle and for generating a signal representative of said speed differences,
means for determining each occurrence when said intra-axle speed difference exceeds a predetermined threshold value,
means for determining the number of such occurrences in a plurality of successive time units and for generating a distribution curve,
means for storing a first set of predetermined values representing a first limiting curve, and
means for comparing said number of occurrences from said distribution curve with said first set of stored predetermined values representing said first limiting curve, and
mode-selecting means for switching said vehicle from said two-wheel to said four-wheel drive mode and for continuously maintaining said vehicle in said four-wheel drive mode when said distribution curve exceeds said first limiting curve, and for switching said vehicle from said four-wheel to said two-wheel drive mode and for continuously maintaining said vehicle in said two wheel drive mode when said first limiting curve exceeds said distribution curve.

2. The vehicle of claim **1** wherein said first limiting curve represents a first set of predetermined values for a predetermined coefficient of road friction.

3. The vehicle of claim 1 comprising
means for storing a second set of predetermined values representing a second limiting curve, and
means for comparing said number of occurrences from said distribution curve with said second set of stored predetermined values representing said second limiting curve,
said mode-selecting means selectively said two-wheel drive mode when said second limiting curve exceeds said distribution curve.

4. The vehicle of claim 3 comprising
a shiftable transmission which is operable in a plurality of selectable gear stages,
means for storing first and second sets of predetermined values representing first and second limiting curves for each selectable gear stage, and
means for comparing said number of occurrences from said distribution curve with said first and second sets of stored predetermined values representing first and second limiting curves for each gear stage,
said mode-selecting means selecting said two-wheel and four-wheel drive modes based upon said comparison for each gear stage.

5. The vehicle of claim **4** further comprising means for detecting a selected gear stage and for generating a signal representative of said selected gear stage, said mode-selecting means being responsive to said signal representing said selected gear stage.

6. The vehicle of claim **3** further comprising means for detecting a steering angle of said vehicle and for generating a signal representative of said steering angle, said mode-selecting means being responsive to said signal representing said steering angle.

7. The vehicle of claim **1** comprising
a shiftable transmission which is operable in a plurality of selectable gear stages,
means for storing a first set of predetermined values representing a first limiting curve for each selected gear stage, and
means for comparing said number of occurrences from said distribution curve with said first set of stored predetermined values representing said first limiting curve for each gear stage,
said mode-selecting means selecting said four-wheel drive when said distribution curve exceeds said first limiting curve for each gear stage.

8. The vehicle of claim **7** further comprising means for detecting a selected gear stage and for generating a signal representative of said selected gear stage, said mode-selecting means being responsive to said signal representing said selected gear stage.

9. The vehicle of claim **1** further comprising means for detecting a steering angle of said vehicle and for generating a signal representative of said steering angle, said mode-selecting means being responsive to said signal representing said steering angle.

10. The vehicle of claim **1** further comprising means for detecting a steering angle of said vehicle and for generating a signal representative of said steering angle, said mode-selecting means maintaining said vehicle in selected two-wheel and four-wheel drive modes when said steering angle signal exceeds a predetermined upper limit.

11. The vehicle of claim **1** wherein said means for generating a distribution curve generates said distribution curve as a function of said number of occurrences detected in successive units of time.

12. The vehicle of claim **1** wherein said means for generating said distribution curve generates said distribution curve as a function of the number of successive units of time in which said occurrences are determined.

13. A motor vehicle comprising
first and second axles,

first and second permanently driven wheels mounted on said first axle,

third and fourth optionally driven wheels mounted on said second axle,

a four-wheel drive for selectively driving only said first and second wheels in a two-wheel drive mode and said first, second, third and fourth wheels in a four-wheel drive mode,

means for detecting intra-axle speed differences between said first and second permanently driven wheels on said first axle and for generating a signal representative of said speed differences,

means for determining each occurrence when said intra-axle speed difference exceeds a predetermined threshold value,

means for determining the number of such occurrences in a plurality of successive time units and for generating a distribution curve,

means for storing first and second sets of predetermined values representing first and second limiting curves respectively,

means for comparing said number of occurrences from said distribution curve with said first and second sets of stored predetermined values representing said first and second limiting curves, and

mode-selecting means for switching said vehicle from said two-wheel to said four-wheel drive mode when said distribution curve exceeds said first limiting curve, and for switching said vehicle from said four-wheel to said two-wheel drive mode when said second limiting curve exceeds said distribution curve.

14. The motor vehicle of claim 13 wherein said first and second sets of predetermined values are equal.

15. A motor vehicle comprising

first and second axles,

first and second permanently driven wheels mounted on said first axle,

third and fourth optionally driven wheels mounted on said second axle,

a four-wheel drive for selectively driving only said first and second wheels in a two-wheel drive mode, and said first, second, third and fourth wheels in a four-wheel drive mode,

means for detecting intra-angle speed differences between said first and second permanently driven wheels on said first axle and for generating a signal representative of said speed differences,

mode-selecting means responsive to said signal for causing said four-wheel drive to selectively operate said motor vehicle in said two-wheel and said four-wheel drive modes,

a shiftable transmission which is operable in a plurality of selectable gear stages,

means for determining each occurrence when said intra-axle speed difference exceeds a predetermined threshold value,

means for determining the number of such occurrences in a plurality of successive time units and for generating a distribution curve,

means for storing first and second sets of predetermined values representing first and second limiting curves for each selectable gear stage, and

means for comparing said number of occurrences from said distribution curve with said first and second sets of stored predetermined values representing first and second limiting curves for each gear stage,

said mode-selecting means selecting said two-wheel and four-wheel drive modes based upon said comparison for each gear stage.

16. The vehicle of claim 15 further comprising means for detecting a selected gear stage and for generating a signal representative of said selected gear stage, said mode-selecting means being responsive to said signal representing said selected gear stage.

17. The vehicle of claim 15 further comprising means for detecting a steering angle of said vehicle and for generating a signal representative of said steering angle, said mode-selecting means being responsive to said signal representing said steering angle.

18. A motor vehicle comprising

first and second axles,

first and second permanently driven wheels mounted on said first axle,

third and fourth optionally driven wheels mounted on said axle,

a four-wheel drive for selectively driving only said first and second wheels in a two-wheel drive mode, and said first, second, third and fourth wheels in a four-wheel drive mode,

means for detecting intra-axle speed differences between said first and second permanently driven wheels on said first axle and for generating a signal representative of said speed differences,

mode-selecting means responsive to said signal for causing said four-wheel drive to selectively operate said motor vehicle in said two-wheel and said four-wheel drive modes,

means for determining each occurrence when said intra-axle speed difference exceeds a predetermined threshold value,

means for determining the number of such occurrences in a plurality of successive time units and for generating a distribution curve,

a shiftable transmission which is operable in a plurality of selectable gear stages,

means for storing a first set of predetermined values representing a first limiting curve for each selected gear stage, and

means for comparing said number of occurrences from said distribution curve with said first set of stored predetermined values representing said first limiting curve for each gear stage,

said mode-selecting means selectively said four-wheel drive mode when said distribution curve exceeds said first limiting curve for each gear stage.

19. The vehicle of claim 18 further comprising means for detecting a selected gear stage and for generating a signal representative of said selected gear stage, said mode-selecting means being responsive to said signal representing said selected gear stage.

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04/05/2004, EAST Version: 1.4.1

United States Patent [19]

Ashauer et al.

[11] **Patent Number: 4,605,087**

[45] **Date of Patent: Aug. 12, 1986**

[54] **ALL-WHEEL DRIVE SYSTEM FOR VEHICLES**

[75] Inventors: **Karl Ashauer**, Wolfsburg; **Bernd Richter**, Bokensdorf; **Manfred Kalversberg**; **Rudiger Schmidt**, both of Wolfsburg, all of Fed. Rep. of Germany

[73] Assignee: **Volkswagenwerk Aktiengesellschaft**, Wolfsburg, Fed. Rep. of Germany

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[51] **Int. Cl.**[4] **B60K 17/344**

[52] **U.S. Cl.** **180/248**; 180/233; 180/244

[58] **Field of Search** 180/233, 247-250, 180/244

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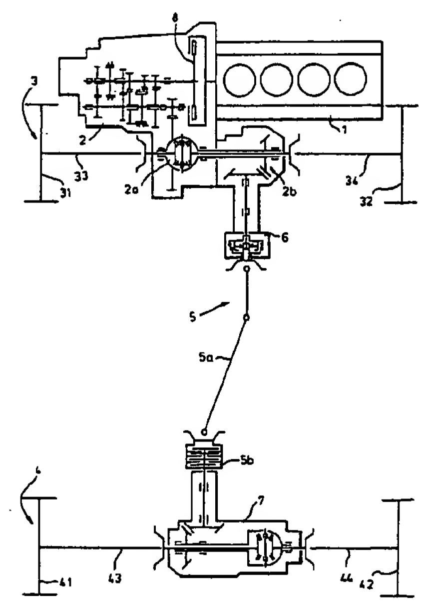
Primary Examiner—Samuel Scott
Assistant Examiner—Noah P. Kamen
Attorney, Agent, or Firm—Brumbaugh, Graves, Donohue & Raymond

[57] **ABSTRACT**

In the particular embodiments described in the specification, an all-wheel drive system is disclosed wherein either the wheels of both the front axle and the rear axle of a vehicle are driven continuously and the front axle and rear axle can be rigidly coupled with each other, or wherein the wheels of only one axle are driven continuously while the drive for the wheels of the second axle can be obtained automatically by means of a viscosity clutch arranged in the driving train between the front axle and the rear axle so that the front and the rear axles are coupled with each other essentially rigidly according to the torque. In addition, one or more coupling devices are provided which are constructed so that a decrease in the rotational speed of the front wheels, caused in particular by the actuation of the service brake, cannot be transmitted through the driving train to the rear wheels.

8 Claims, 4 Drawing Figures



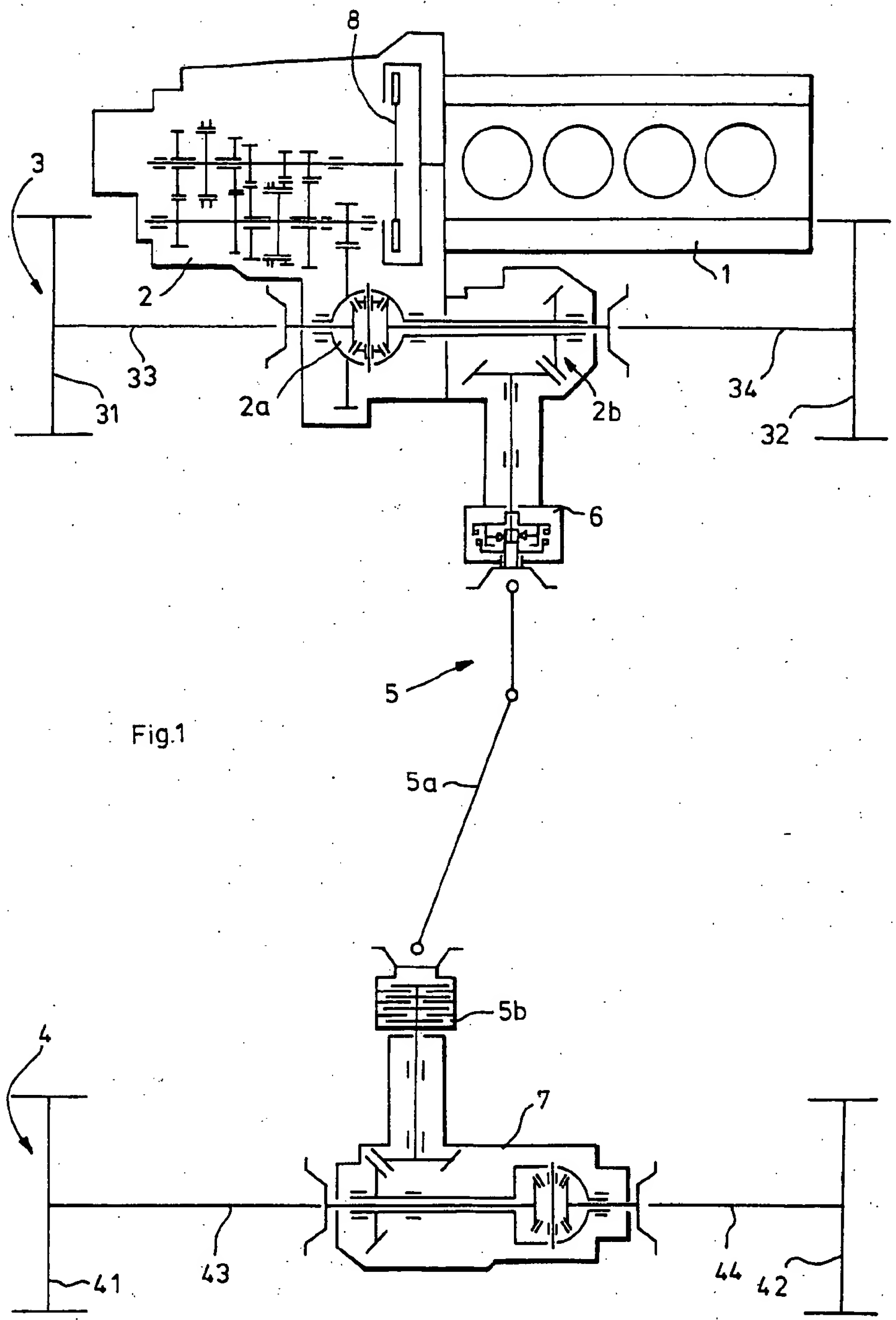


Fig.1

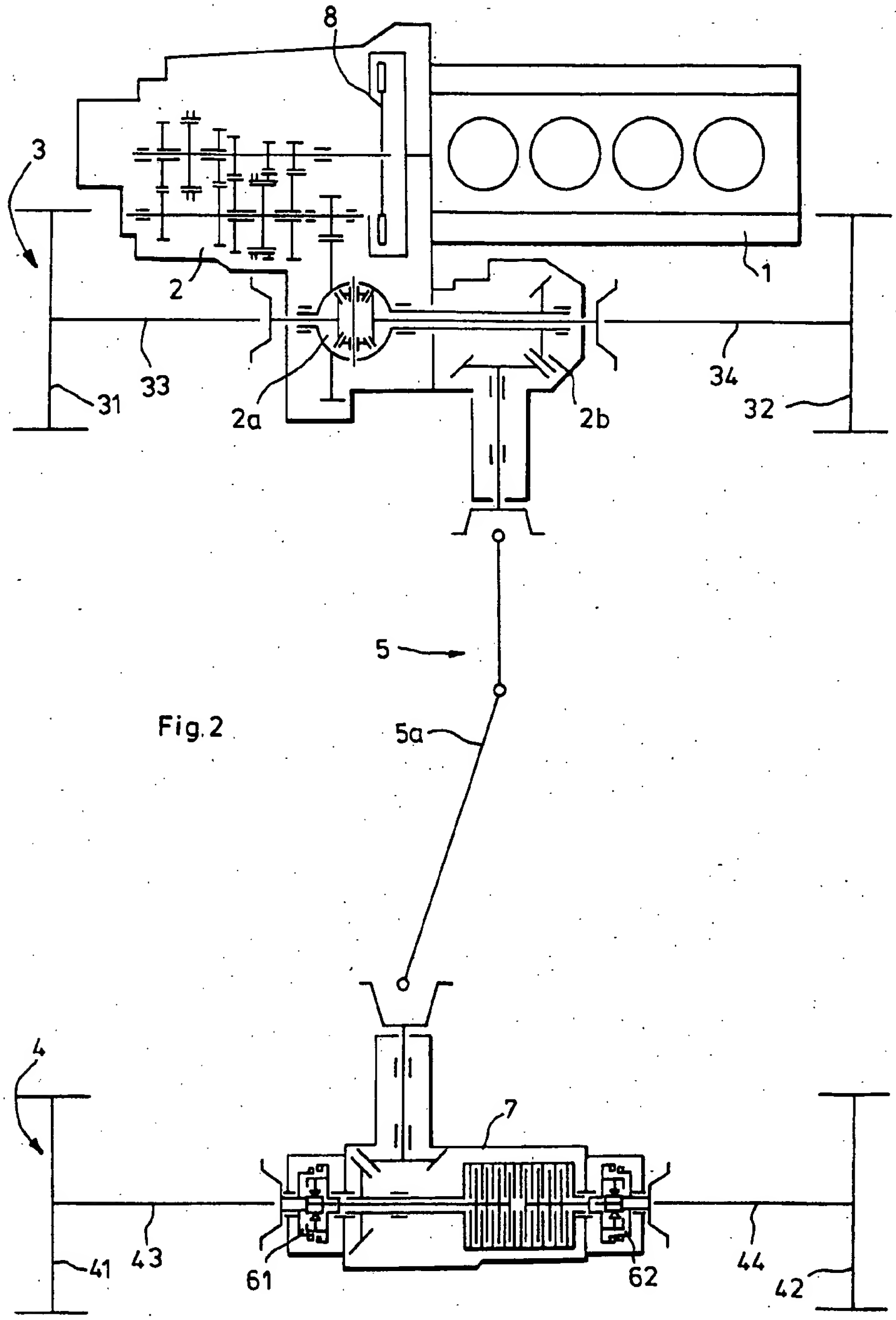


Fig. 2

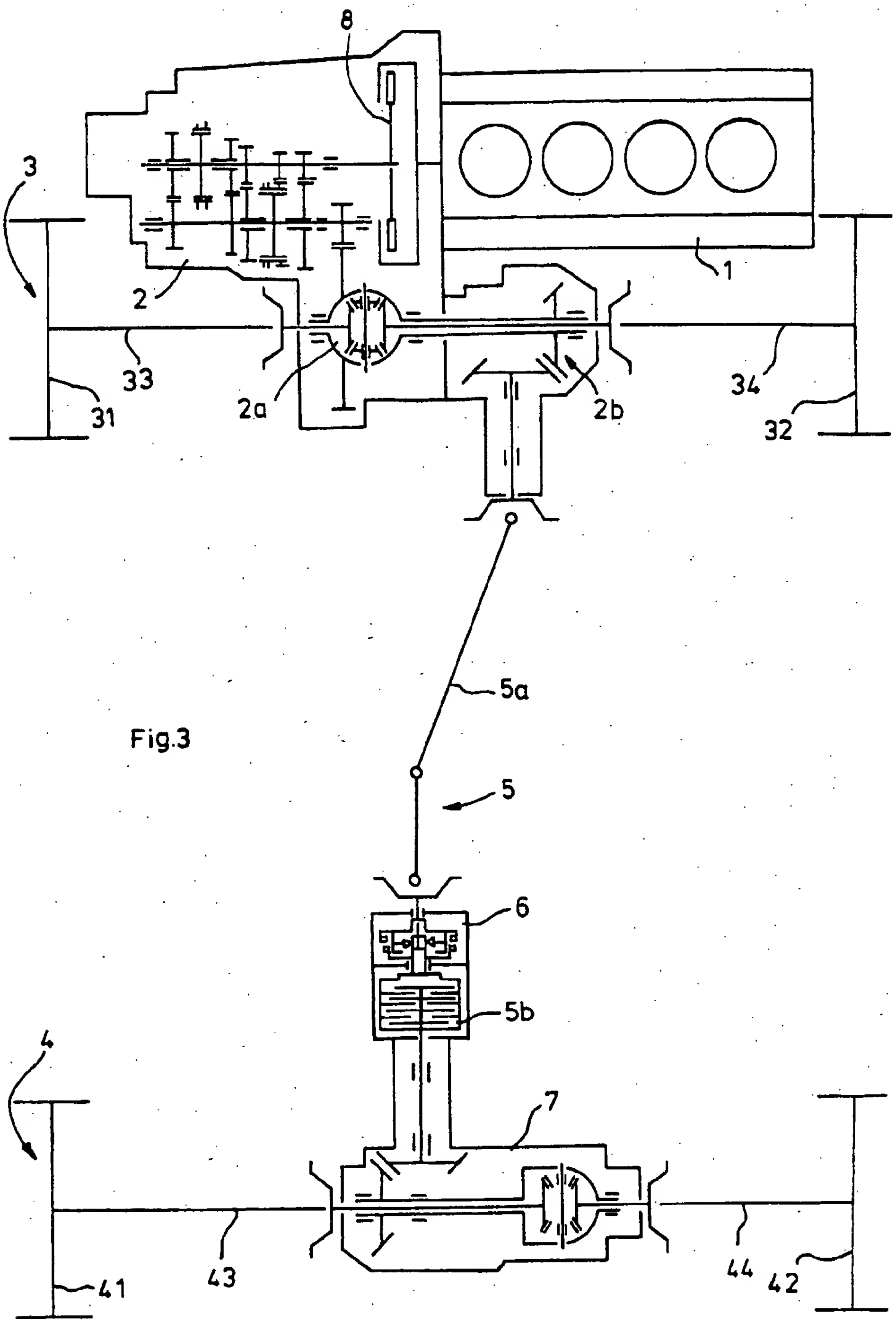


Fig.3

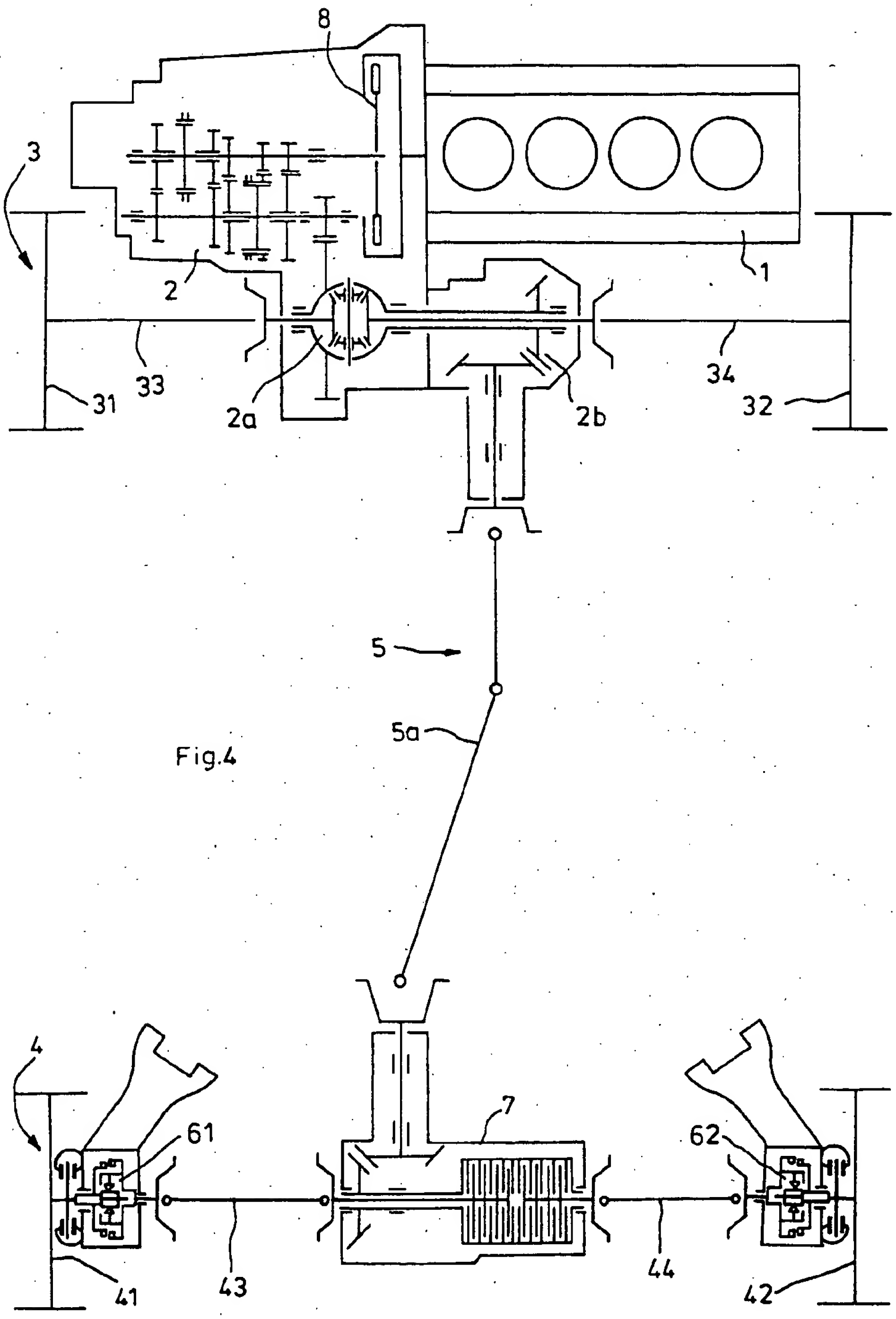


Fig.4

ALL-WHEEL DRIVE SYSTEM FOR VEHICLES

BACKGROUND OF THE INVENTION

The invention relates to all-wheel drive systems for vehicles and, more particularly, to a new and improved all-wheel drive system having a controllable coupling between the front and rear wheels of the vehicle.

All-wheel drive systems for vehicles can be roughly subdivided into two groups, namely, one in which the four wheels are driven permanently, i.e., continuously, independent of road conditions (e.g., the Audi Quattro and Audi 80 Quattro), and one in which the wheels of one axle are driven continuously and the wheels of the other axle are driven only at certain times such as during cross country driving. In the latter case, the connection of the second axle drive occurs either automatically by means of a mechanical overrunning device when a predetermined slippage of the steadily driven axle is exceeded (e.g., German Pat. No. 892,275; DE-AS No. 1,077,537; DE-OS No. 2,413 288; DE-OS No. 2,928,351), or it is initiated manually by the operator whenever necessary by actuation of a corresponding control lever.

A driving arrangement (e.g., EP OS No. 68,309) whereby the wheels of one axle only are continuously driven by way of a conventional axle drive while the drive for the wheels of the second axle is obtained automatically by means of a viscosity coupling arranged in the driving train between the front and the rear axles corresponds closely to a permanent all-wheel drive system.

Such viscosity couplings transmit only very small torques at small speed differences but transmit very large torques at higher speed differences. Depending on the point of installation within the driving train, such viscosity couplings can therefore assume either the tasks of only a center differential or the tasks of both a center differential and an axle differential. Especially under low traction conditions, which occur when roads are wet, icy, snow-covered or sandy, an all-wheel drive with a viscosity coupling acts, depending on the point of installation, either as a permanent all-wheel drive with a blocked center differential or as permanent all-wheel drive with a blocked center differential and a blocked axle differential.

In general, such a rigid coupling between the driven front and rear wheels provides certain advantages on braking, including shortening of the braking distance. When the front wheels of a vehicle are overbraked, resulting in locking, however, coupling of the two axles also causes the rear wheels to be over-braked and locked through the driving train between the front wheels and the rear wheels. This occurs even though the braking system of a vehicle such as an automobile is usually arranged so that the braking effect is larger on the front axle than on the rear axle, assuring that the front wheels will lock before the maximum possible braking forces are attained on the rear wheels. It is known that locking of the rear wheels leads to loss of driving stability.

Accordingly, it is an object of the present invention to provide an improved all-wheel drive system for vehicles having driving wheels which are connected permanently and, in particular, to provide such a system in which the driving stability of the vehicle is not impaired even on overbraking of the front wheels.

SUMMARY OF THE INVENTION

In accordance with the invention, an all-wheel drive system for vehicles includes a device for uncoupling the drive train between the front and rear wheels of the vehicle in accordance with the applied torque, such that a decrease in the rotational speed of the front wheels of the vehicle can not be transmitted to the rear wheels through the driving connection between them. Accordingly, when the service brake is actuated with a rigid coupling between the front and the rear wheels, the system of the invention responds to the braking action on the front wheels so as to uncouple the rear wheels so that any overbraking of the front wheels can not be transmitted to the rear wheels through the driving train.

Although some automobiles with all-wheel drive include a device between the front and the rear wheels that can be coupled and uncoupled so that a decrease in the rotational speed of the front wheels is not transmitted to the rear wheels through the driving train, they do not correspond to the system of the present invention.

For example, a known automobile of this kind (DE-OS No. 2,413,288) has a constantly driven front axle and a rear axle which becomes coupled automatically, by means of a blockable mechanical overrunning device, when the front axle slippage reaches about 5 to 15%. Basically, this known automobile is a conventional front-driven automobile which has an all-wheel drive only when it is driven under unfavorable conditions, in particular rough terrain. Under those conditions the overrunning device merely assumes the task of the manual shift lever, normally actuated by the operator, for connecting in the second axle to the driven axle. Thus it is not a permanent all-wheel drive system. In case of known automobiles with permanent all-wheel drives in which both the front and the rear wheels are driven even during normal operation, there is no overrunning device since it is not required for all-wheel drive operation. It would therefore, seem inappropriate to those skilled in the art to provide such permanent all-wheel drive vehicles with an uncoupling device and it would not be apparent to do so in order to prevent a decrease in the rotational speed of the front wheels, in particular, one caused by the service brake, to be transmitted through the driving train to the rear wheels. Such an arrangement therefore is contrary to the teachings of the art.

All-wheel drive automobiles are also known (DE-OS No. 3,127,605) wherein the front and the rear wheels are normally driven constantly and wherein an engageable and disengageable clutch is inserted between the driving axles of the front and the rear wheels. These automobiles are intended only for racing and rally operations. The clutch which is provided serves to enable the operator to disengage the front wheel drive during operation so that subsequently, by actuation of the hand brake, he may lock the rear wheels when initiating a narrow curve, such as is customary in racing and rallye operation. This known all-wheel concept could not contribute anything to the solution of the problem solved by the invention, for it is precisely the aim of that arrangement to permit what is to be prevented by the subject of the application, namely, the locking of the rear wheels.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will be explained in detail in the following description with reference to the drawings, in which:

FIG. 1 is a schematic diagram illustrating one embodiment of an all-wheel drive system according to the invention;

FIG. 2 is a similar schematic diagram illustrating another embodiment of the drive system of the invention;

FIG. 3 Illustrates schematically a further embodiment of the drive system of the invention in which the coupling means is located at the output from the universal shaft to the rear axle; and

FIG. 4 is a schematic diagram of a further embodiment of the drive system of the invention in which the coupling means is located adjacent the rear wheels.

DESCRIPTION OF PREFERRED EMBODIMENTS

In the representative embodiment of the invention shown in FIG. 1, a driving engine 1 having a clutch 8 for connection to a transmission 2 is arranged in the front of a vehicle so that the wheels 31 and 32 of the front axle 3 are driven through a differential 2a and corresponding shafts 33 and 34 whenever the clutch and transmission are engaged.

The rear axle 4 of the vehicle has wheels 41 and 42. These are driven by means of a driving train 5 coupled to the front axle differential 2a by means of a bevel gear arrangement 2b and similarly to a rear axle differential 7 which is connected to the rear wheels 41 and 42 through corresponding drive shafts 43 and 44. Within the driving train there is a viscosity coupling 5b located at the input of the rear axle differential 7. Such a viscosity coupling has the property of transmitting only small torques if the speed differences are small between its input and output shaft. However, it is able to transmit large torques if larger speed differences between the input and the output occur. Thus, when the steadily and directly driven front wheels 31 and 32 rotate with very small slippage due to very good road conditions, i.e., when the rotational speed of the front and the rear wheels is at least approximately equal, the viscosity coupling will transmit practically no torque and thus, the rear axle will follow the front axle with no significant driving action. Under different conditions, in particular on wet, snow-covered or icy roads, for travel on which it is recommended that the center differential of an automobile with a permanent four-wheel drive be blocked, the viscosity coupling 5b, in contrast to the good road conditions, will produce a rigid, torque-transmitting connection between its input and output. Under such operating conditions, locking of the front wheels due to overbraking would therefore normally result in a corresponding locking of the rear wheels by way of the driving train 5.

In accordance with the invention, a device 6 is provided for coupling and uncoupling the driving train 5 based on the torque transmitted between the front axle 3 and the rear axle 4. The device 6 is constructed so that a decrease in the rotational speed of the front wheels 31 and 32, caused in particular by actuation of the service brake, can not be transmitted to the rear wheels 41 and 42 via the driving train 5. In the representative embodiment shown in FIG. 1, the device 6 is arranged at the input to a universal shaft 5a leading to the rear axle. It is apparent that the device 6 can also be located at any other point in the driving train 5, such as at the output of the universal shaft 5a at the rear axle differential 7.

In the illustrated embodiment, the device 6 constitutes an overrunning device which is automatically coupled and uncoupled according to the direction of the torque, e.g., such as a grip roller overrunning device.

Because of the presence of the overrunning device 6 in the otherwise permanent all-wheel drive system, the possibility that locking of the front wheels 31 and 32 could cause locking of the rear wheels 41 and 42 through the driving train 5 is eliminated since the overrunning device automatically interrupts the transmission of torque in this direction through the drive train.

In order to facilitate an all-wheel drive even when the vehicle is driven backward, the overrunning device 6 is preferably designed so that it can be blocked, either manually or automatically, when the reverse gear is engaged.

As an alternative to the illustrated device 6, it is also possible to use a controllable clutch which is disengaged automatically on actuation of the service brake (foot brake). This may be accomplished, for example, by a control signal, generated by actuation of the service brake, which causes actuation of the clutch in a known manner, e.g., either hydraulically or electromagnetically.

Instead of a single coupling device 6 located in the drive train 5 a similar coupling device may be located in each of the driving trains 43 and 44 leading from the rear axle differential 7 to the rear wheels 41 and 42. One such arrangement is illustrated in FIG. 2 in which two devices 61 and 62, similar to the device 6 of FIG. 1, are arranged at the input of the two drive shafts 43 and 44 of the rear axle differential 7. Obviously, it is also possible to arrange these devices at the output of the drive shaft 43 and 44 adjacent to the rear wheels 41 and 42.

As explained with reference to the device 6 in the embodiment illustrated in FIG. 1, the devices 61 and 62 may be constructed either as overrunning devices capable of being coupled and uncoupled automatically or as controllable clutches.

Use of two overrunning devices or clutches in the driving trains leading from the rear axle differential to the rear wheels is of special advantage in case the rear axle differential is constructed in the manner of a known limited slip differential or torque splitter (EP-OS No. 68,309) which operates as an automatically blockable center differential as well as an automatically controllable rear axle differential and permits only small speed differences between the two wheels of the said axle. Utilization of the devices 61 and 62 in such vehicles not only ensures that a decrease in the rotational speed, and thereby also a locking, of the front wheels, caused in particular by actuation of the service brake, is not transferred to the rear wheels via the driving train, it also prevents locking of one of the wheels of the rear axle running on a surface with low traction from causing premature over-braking of the other wheel of the rear axle which is running on a surface with a high traction. If, for example, the wheels 41 and 42 were rigidly connected with each other by means of a blockable differential or the like, the wheel running on a good surface would be braked not only by its own wheel brake but also the brake of the prematurely locked wheel through the rigid drive shaft coupling.

Although the invention has been described herein with reference to specific embodiments, many modifications and variations therein will readily occur to those skilled in the art. Accordingly, all such variations are included within the intended scope of the invention as defined by the following claims.

We claim:

1. In an all-wheel drive system for a vehicle comprising a front axle with front wheels and a rear axle with rear wheels, the front wheels being continuously driven by a conventional live axle drive comprising a positive differential gear and the rear wheels being automatically driven with said front wheels by a drive train having viscous fluid coupling means therein, the improvement comprising at least one coupling means in said drive train responsive to the direction of a torque applied thereto to automatically interrupt said drive train so as to prevent braking effects on the front wheels from being transmitted to the rear wheels by way of the drive train.

2. An all-wheel drive system according to claim 1 wherein the drive train includes a universal shaft leading to the rear axle and having opposite ends near and away from the rear axle, respectively, and the coupling means is interposed in the drive train at one of said ends of the universal shaft.

3. An all-wheel drive system according to claim 2 wherein the coupling means is interposed in the drive train at the end of the universal shaft away from the rear axle.

4. An all-wheel drive system according to claim 2 wherein the coupling means is interposed in the drive train at the end of the universal shaft near the rear axle.

5. An all-wheel drive system according to claim 1 wherein the drive train includes a universal shaft leading to the rear axle and two drive shafts leading, respectively, to each of the rear wheels and each having an end near and an end away from its respective wheel and wherein said coupling means comprises coupling elements interposed in said drive train at one end of each of said drive shafts, said coupling elements being responsive to the direction of torque applied thereto for interrupting said drive train so as to prevent braking effects on the front wheels from being transmitted to the rear wheels by way of the drive train.

6. An all-wheel drive system according to claim 5 wherein said respective coupling elements are interposed in the drive train at the ends of the respective drive shafts away from the rear wheels.

7. An all-wheel drive system according to claim 5 wherein said respective coupling elements are interposed in the drive train at the ends of the respective drive shafts adjacent to the rear wheels.

8. An all-wheel drive system according to any of claims 1-7, wherein the coupling means comprises overrunning means arranged to be automatically coupled and uncoupled in response to the direction of the torque applied thereto.

* * * * *

United States Patent [19]

Hayashi

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[45] Date of Patent: Oct. 21, 1986

[54] POWER TRANSFER DEVICE FOR FOUR-WHEEL DRIVE VEHICLE

[75] Inventor: Hideyuki Hayashi, Toyota, Japan
[73] Assignee: Toyota Jidosha Kabushiki Kaisha, Japan

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[52] U.S. Cl. 180/249; 180/233; 180/248
[58] Field of Search 180/249, 248, 247, 233; 74/710.5, 713, 714, 740, 748

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Primary Examiner—John J. Love
Assistant Examiner—Donn McGiehan
Attorney, Agent, or Firm—Parkhurst & Oliff

[57] ABSTRACT

A power transfer device for a four-wheel drive vehicle comprises a first differential including a differential case arranged to be applied with drive torque from a transmission, a pinion gear rotatably mounted within the differential case, and a pair of side gears rotatably mounted within the differential case and in mesh with the pinion gear; a pair of output shafts connected to the side gears respectively for driving the front and rear road wheels; a second differential including a differential carrier formed to contain the differential case therein, a carrier rotatably mounted within the differential carrier and connected with the differential case for rotation therewith about a common axis, a sun gear integral with one of the side gears, a planetary gear rotatably supported by the carrier and in mesh with the sun gear, a ring gear integral with the inner wall of the differential carrier and in mesh with the planetary gear, the carrier being drivingly connected to an output shaft of the transmission; and a selector mechanism arranged to drivingly connect one of the output shafts or the differential carrier to the front or rear road wheels.

7 Claims, 7 Drawing Figures

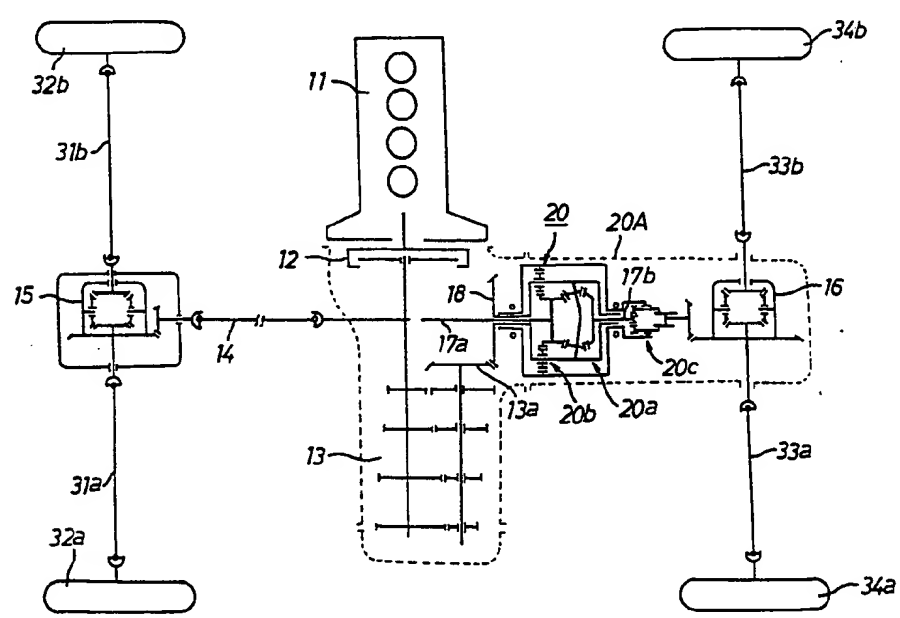
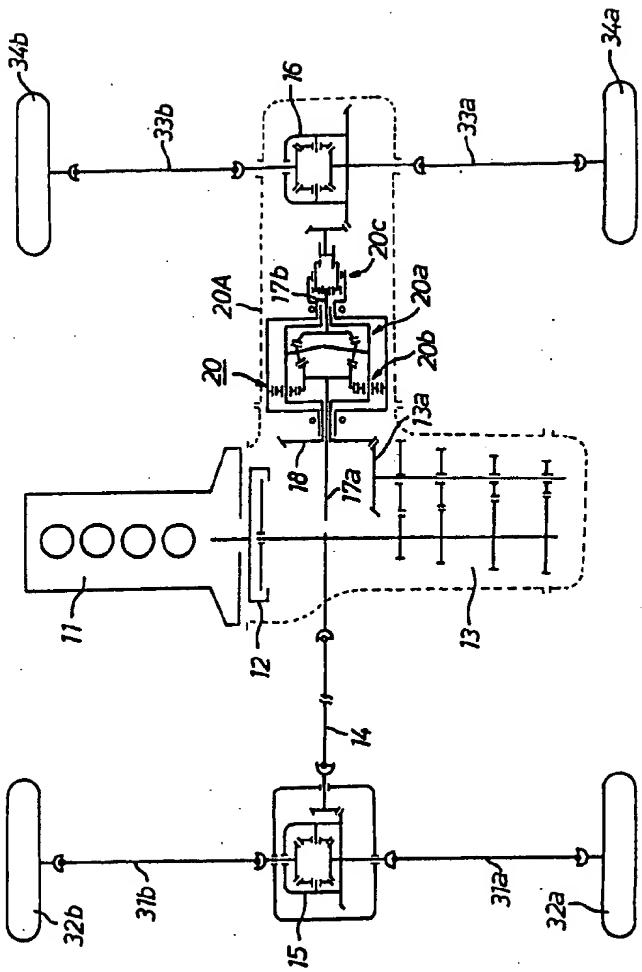


Fig. 1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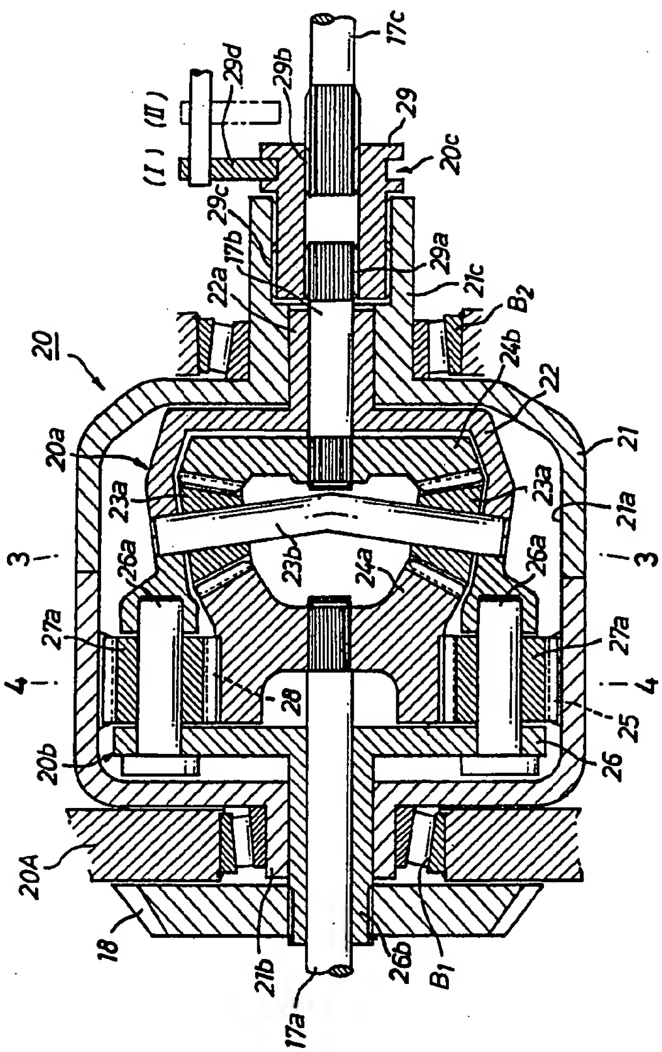
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Fig. 2

Fig. 3

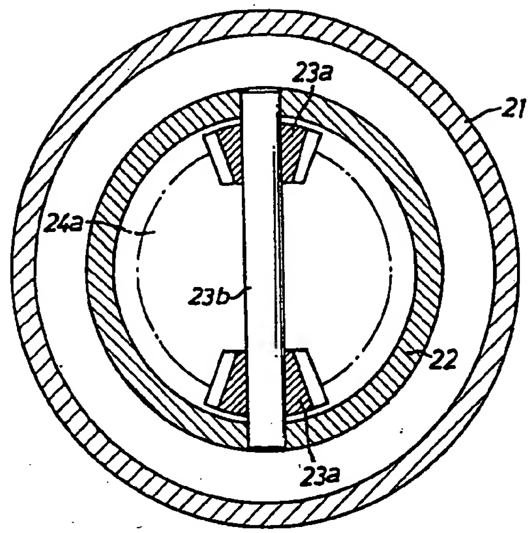
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Fig. 4

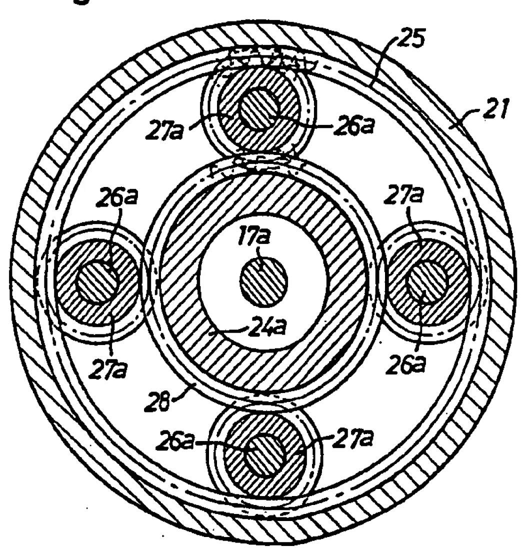
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Fig. 6

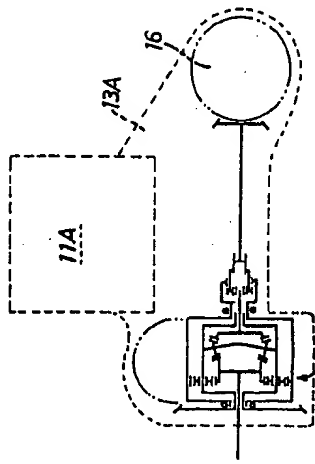


Fig. 7

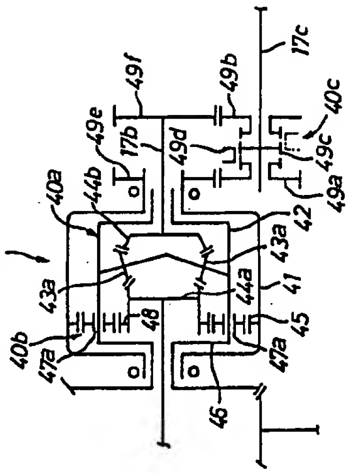
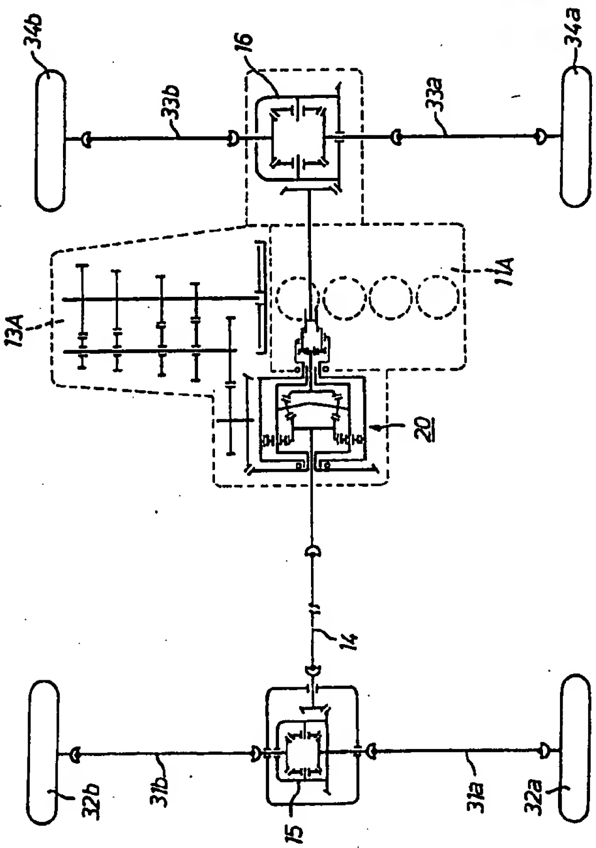


Fig. 5



POWER TRANSFER DEVICE FOR FOUR-WHEEL DRIVE VEHICLE

BACKGROUND OF THE INVENTION

1. Field of the invention

The present invention relates to a power transfer device for use in a four-wheel drive vehicle, more particularly to a power transfer device capable of controlling the split ratio of the drive torque to the front and rear road wheels in accordance with driving conditions of the vehicle.

2. Discussion of the background

In conventional power transfer devices for four-wheel drive, a differential of the bevel gear type or the planetary gear type has been adapted to transfer drive torque from a prime mover to the front and rear road wheels at a constant gear ratio. It is, however, noted that the steering stability of the vehicle is in a close relationship with the split drive torque applied to the front and rear road wheels. For this reason, it is desirable that the split ratio of the drive torque to the front and rear road wheels is controlled in accordance with road conditions and driving conditions such as frictional coefficient of the road surface, inclination of the road, acceleration of the vehicle, cornering travel of the vehicle and the like. For such control of the split ratio of the drive torque, the Japanese Patent Early Publication No. 59 - 151661 discloses a power transfer device wherein a continuously variable transmission is disposed in a front or rear wheel drive power train of the differential. In this power transfer device, the change-speed ratio of the transmission is changed in accordance with driving conditions of the vehicle to control the split ratio of the drive torque to the front and rear road wheels. It is, however, disadvantageous that the power transfer device becomes large in size and heavy due to provision of the continuously variable transmission.

SUMMARY OF THE INVENTION

It is, therefore, a primary object of the present invention to provide a compact power transfer device capable of controlling the split ratio of the drive torque to the front and rear road wheels in accordance with driving conditions of the vehicle.

According to the present invention there is provided a power transfer device for use in a four-wheel drive vehicle, wherein a first differential of the bevel gear type is associated with a second differential of the planetary gear type about a common axis to transfer drive torque from a transmission to the front and rear road wheels at a desired gear ratio. The first differential includes a differential case arranged to be applied with drive torque from the transmission, a pair of pinion gears rotatably mounted within the differential case, and a pair of side gears rotatably mounted within the differential case and in mesh with the pinion gears. A first output shaft is connected to one of the side gears for driving the front or rear road wheels, and a second output shaft is connected to the other side gear for driving the rear or front road wheels. The second differential includes a differential carrier formed to contain the differential case therein, a carrier rotatably mounted within the differential carrier and connected with the differential case for rotation therewith about a common axis, a sun gear integral with the side gear connected to the first output shaft, a plurality of planetary gears rotatably supported by the carrier and in mesh with the sun gear, a ring gear integral with the inner wall of the differential carrier and in mesh with the planetary gears. The carrier is drivingly connected to an output shaft of the transmission. A selector mechanism is arranged to drivingly connect the second output shaft or the differential carrier to the rear or front road wheels.

The power transfer device is characterized by arrangement of the sun gear integral with the side gear, the ring gear integral with the inner wall of the differential carrier, and the differential case integrally with the carrier. With such arrangement, the first and second differentials can be combined with each other in a limited space. In operation, the carrier is applied with output drive torque from the transmission, and in turn, the differential case is applied with the drive torque from the carrier to drive the side gears through the pinion gears. Thus, the first and second output shafts are applied with the drive torque at a gear ratio of the side gears. When the second output shaft is drivingly connected to the rear or front road wheels under control of the selector mechanism, the drive torque is transmitted to the front and rear road wheels at the gear ratio of the side gears. When the differential carrier is drivingly connected to the rear or front road wheels under control of the selector mechanism, the planetary gears act to transfer the drive torque from the carrier to the sun gear and the ring gear. The split drive torque to the sun gear is transmitted to the front or rear road wheels through the first output shaft, while the split drive torque to the ring gear is transmitted to the rear or front road wheels through the differential carrier. In this instance, the split ratio of the drive torque to the front and rear road wheels is determined by a gear ratio of the sun gear and the ring gear.

BRIEF DESCRIPTION OF THE DRAWINGS

Other objects, features and advantages of the present invention will become readily apparent from the following detailed description of preferred embodiments thereof when taken in connection with the accompanying drawings, in which:

FIG. 1 is a schematic illustration of a four-wheel drive vehicle equipped with a power transfer device in accordance with the present invention;

FIG. 2 is a sectioned plan view of the power transfer device shown in FIG. 1;

FIG. 3 illustrates a cross-section taken along lines 3—3 in FIG. 2;

FIG. 4 illustrates a cross-section taken along lines 4—4 in FIG. 2;

FIG. 5 is a schematic illustration of arrangement of the power transfer device in another four-wheel drive vehicle;

FIG. 6 is a side view showing the arrangement of the power transfer device in relation to an internal combustion engine and a rear differential shown in FIG. 5; and

FIG. 7 is a schematic illustration of a modification of the power transfer device.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings and particularly to FIG. 1, there is schematically illustrated a four-wheel drive vehicle of the midship type, comprising an internal combustion engine 11 mounted on a vehicle chassis (not shown) in a transverse direction, a transmission 13 drivingly connected to the engine 11 through a clutch

assembly 12, and a power transfer device 20 arranged in a fore-and-aft direction of the vehicle and drivingly connected to an output bevel gear 13*a* of the transmission 13. The power transfer device 20 has an input bevel gear 18 in mesh with the output bevel gear 13*a* of transmission 13, a first output shaft 17*a* drivingly connected to a front propeller shaft 14 which in turn is drivingly connected to a front differential 15, and a second output shaft 17*b* drivingly connected through a selector mechanism 20*c* to a rear differential 16. The front differential 15 is arranged to drive a set of front road wheels 32*a* and 32*b* through front split axle parts 31*a* and 31*b*, and the rear differential 16 is arranged to drive a set of rear road wheels 34*a* and 34*b* through rear split axle parts 33*a* and 33*b*.

As shown in FIGS. 2 to 4, the power transfer device 20 comprises a first differential 20*a* of the bevel gear type and a second differential 20*b* of the planetary gear type which are contained within a differential carrier 21 for the second differential 20*b*. The first differential 20*a* includes a differential case 22 rotatably mounted within the differential carrier 21, a pair of pinion gears 23*a*, and a pair of side gears 24*a* and 24*b*. The second differential 20*b* includes a ring gear 25 integral with the inner wall 21*a* of differential carrier 21, a carrier 26 rotatably mounted within the differential carrier 21, a plurality of planetary gears 27*a* in mesh with the ring gear 25, and a sun gear 28 integral with the side gear 24*a* and in mesh with the planetary gears 27*a*. The differential carrier 21 is contained within a trans-axle casing 20A for the transfer device 20 and has a pair of axially spaced sleeve portions 21*b* and 21*c* which are rotatably supported by a pair of axially spaced bearings B_1 and B_2 on the trans-axle casing 20A.

The differential case 22 is integrally connected with the carrier 26 by means of a plurality of circumferentially equi-spaced support pins 26*a* on which the planetary gears 27*a* are rotatably mounted, respectively. (see FIG. 4) the differential case 22 has a sleeve portion 22*a* rotatably coupled within the sleeve portion 21*c* of differential carrier 21, and the carrier 26 has a sleeve portion 26*b* rotatably coupled within the sleeve portion 21*b* of differential carrier 21. The input bevel gear 18 is splined to the outer end of sleeve portion 26*b* for rotation therewith. The pinion gears 23*a* each are rotatably mounted on a cross shaft 23*b* which is carried on the differential case 22. The pinion gears 23*a* are in mesh with the side gears 24*a* and 24*b* which are splined to the inner ends of output shafts 17*a* and 17*b*, respectively. The side gear 24*a* is formed smaller in diameter than the side gear 24*b* and has a sleeve portion formed with the sun gear 28. The first output shaft 17*a* extends outwardly through the sleeve portion 26*b* of carrier 26, while the second output shaft 17*b* extends outwardly through the sleeve portion 22*a* of differential case 22.

The selector mechanism 20*c* is remotely operated by the vehicle driver to selectively connect the second output shaft 17*b* or the differential carrier 21 to the rear differential 16 through an intermediate drive shaft 17*c*. The selector mechanism 20*c* includes a coupling sleeve 29 which is slidably mounted on the output shaft 17*b* and drive shaft 17*c* for rotation therewith. The coupling sleeve 29 has a first internally splined portion 29*a* slidably engaged with an externally splined portion of output shaft 17*b*, a second internally splined portion 29*b* slidably engaged with an externally splined portion of drive shaft 17*c*, and an externally splined portion 29*c* engageable with an internally splined portion of the sleeve portion 21*c* of differential carrier 21. A remotely operated shift fork 29*d* is engaged with the coupling sleeve 29 to shift it between first and second positions I and II. When retained in the first position I, the coupling sleeve 29 provides a drive connection between the output shaft 17*b* and the drive shaft 17*c*. When shifted to the second position II, the coupling sleeve 29 provides a drive connection between the differential carrier 21 and the drive shaft 17*c*.

In operation of the power transfer device 20, the planetary carrier 26 is applied with output drive torque from the transmission 13 through a final speed reduction gear train including the intermeshed bevel gears 13*a* and 18, and in turn, the differential case 22 is applied with the drive torque from the carrier 26 to drive the side gears 24*a* and 24*b* through the pinion gears 23*a*. Thus, the first and second output shafts 17*a* and 17*b* are applied with the drive torque at a gear ratio of the side gears 24*a* and 24*b*. Assuming that the coupling sleeve 29 of selector mechanism 20*c* is retained in the first position I, the split drive torque to the first output shaft 17*a* is transmitted to the front road wheels 32*a* and 32*b* by way of the front propeller shaft 14, front differential 15, and split axle parts 31*a* and 31*b*, while the split drive torque to the second output shaft 17*b* is transmitted to the rear road wheels 34*a* and 34*b* through the coupling sleeve 29, drive shaft 17*c*, rear differential 16, and split axle parts 33*a* and 33*b*.

When the coupling sleeve 29 is shifted to the second position II to connect the differential carrier 21 to the drive shaft 17*c*, the planetary gears 27*a* act to transfer the drive torque from the carrier 26 to the sun gear 28 and the ring gear 25. The split drive torque to the sun gear 28 is transmitted to the front road wheels 32*a* and 32*b* by way of the side gear 24*a*, first output shaft 17*a*, front propeller shaft 14, differential 15, and split axle parts 31*a* and 31*b*, while the split drive torque to the ring gear 25 is transmitted to the rear road wheels 34*a* and 34*b* through the differential carrier 21, coupling sleeve 29, drive shaft 17*c*, rear differential 16, and split axle parts 33*a* and 33*b*. In this instance, the split ratio of the drive torque to the front and rear road wheels is determined by a gear ratio of the ring gear 25 and sun gear 28. Consequently, the split ratio of the drive torque to the front and rear road wheels is selectively determined by operation of the selector mechanism 20*c*.

In the case that the gear ratio of the side gears 24*a* and 24*b* is determined to be 45:55 and that the gear ratio of the sun gear 28 and the ring gear 25 is determined to be 30:70, the split ratio of the drive torque to the front and rear road wheels can be selected in accordance with driving conditions of the vehicle as follows. When the selector mechanism 20*c* is operated to retain the coupling sleeve 29 in the first position I during straight travel of the vehicle, the drive torque is transmitted to the front and rear road wheels at the ratio of 45:55 to enhance acceleration performance of the vehicle. When the selector mechanism 20*c* is operated to shift the coupling sleeve 29 to the second position II during cornering travel of the vehicle, the drive torque is transmitted to the front and rear road wheels at the ratio of 30:70 to enhance cornering performance of the vehicle.

In the case that the gear ratio of the side gears 24*a* and 24*b* is determined to be 50:50 and that the gear ratio of the sun gear 28 and the ring gear 25 is determined to be 30:70, the split ratio of the drive torque to the front and rear road wheels can be selected in accordance with driving conditions of the vehicle as follows. When the

selector mechanism **20c** is operated to retain the coupling sleeve **29** in the first position I during travel of the vehicle on a slippery road such as a snowcovered or frozen road, the drive torque is transmitted to the front and rear road wheels at the ratio of 50:50 to avoid unexpected slip of the vehicle. When the selector mechanism **20c** is operated to shift the coupling sleeve **29** to the second position II during travel of the vehicle on a dry asphalt road, the drive torque is transmitted to the front and rear road wheels at the ratio of 30:70 to enhance drivablity of the vehicle.

From the above description, it will be understood that the power transfer device **20** is characterized by arrangement of the sun gear **28** integral with the sleeve portion of side gear **24a**, the ring gear **25** integral with the inner wall **21a** of differential carrier **21**, and the differential case **22** integral with the carrier **26**. With such arrangement of the sun gear **28**, ring gear **25** and differential case **22**, the first and second differentials **20a** and **20b** can be combined with each other in a limited space. It is, therefore, able to provide the power transfer device **20** in a compact construction.

In FIGS. **5** and **6**, there is schematically illustrated an arrangement of the power transfer device **20** in another four wheel drive vehicle of the midship type, wherein an internal combustion engine **11A** is arranged at the left side of the vehicle and a transmission **13A** is arrnaged at the right side of the vehicle and wherein the power transfer device **20** is drivingly connected to the transmission **13A** at the front side of the engine **11A**.

In FIG. **7**, there is schematically illustrated a modification of the power transfer device **20**, wherein a first differential **40a** of the bevel gear type is integrally combined with a second differential **40b** of the planetary gear type. The first differential **40a** is mounted within a differential carrier **41** for the second differential **40b** and includes a differential case **42**, a pair of pinion gears **43a**, and a pair of side gears **44a** and **44b** which correspond with the components of the first differential **20a** in the power transfer device **20**. The second differential **40b** includes a ring gear **45**, a carrier **46**, a plurality of planetary gears **47a** and a sun gear **48** which correspond with the components of the second differential **20b** in the power transfer device **20**. In the modified power transfer device **40**, the intermediate drive shaft **17c** is arranged in parallel with the second output shaft **17b**. A selector mechanism **40c** associated with the transfer device **40** includes a pair of axially spaced gears **49a** and **49b** rotatably mounted on the drive shaft **17c**, an intermediate hub gear **49c** fixed to the drive shaft **17c**, and a coupling sleeve **49d** slidably engaged with the hub gear **49c** and shiftable between a first position where it couples the gears **49a** and **49c** and a second position where it couples the gears **49b** and **49c**. The gear **49a** is permanently in mesh with a drive gear **49e** integral with the differential carrier **41**, and the gear **49b** is permanently in mesh with a drive gear **49f** integral with the output shaft **17b**.

In the above modification, only the selector mechanism **40c** is different in its construction and arrangement from the selector mechanism **20c** in the power transfer device **20**, wherein under control of the coupling sleeve **49d** the same function as that of the power transfer device **20** is effected to provide the same effects or advantages. The foregoing disclosure is the best mode devised by the inventor for practicing this invention. It is apparent, however, that devices incorporating other modifications and variations to the instant invention will become obvious to one skilled in the art of four wheel drive systems. Inasmuch as the foregoing disclosure is intended to enable one skilled in the pertinent art to practice the instant invention, it should not be construed to be limited thereby but should be construed to include such aforementioned obvious variations and be limited only by the spirit and scope of the following claims.

The embodiments of the invention in which an exclusive property or privilege is claimed are defined as follows.

1. A power transfer device for a four-wheel drive vehicle having a transmission drivingly connected to a prime mover of the vehicle and having a set of front road wheels and a set of rear road wheels adapted to be driven from said transmission, said transfer device comprising:
   - a first differential of the bevel gear type including a differential case arranged to be applied with drive torque from said transmission, a pinion gear rotatably mounted within said differential case, and a pair of side gears rotatably mounted within said differential case and in mesh with said pinion gear;
   - a first output shaft connected to one of said side gears for driving the front or rear road wheels;
   - a second output shaft connected to the other side gear for driving the rear or front road wheels;
   - a second differential of the planetary gear type including a differential carrier formed to contain said differential case therein, a carrier rotatably mounted within said differential carrier and connected with said differential case for rotation therewith about a common axis, a sun gear integral with said side gear connected to the first output shaft, a planetary gear rotatably supported by said carrier and in mesh with said sun gear, a ring gear integral with the inner wall of said differential carrier and in mesh with said planetary gear, said carrier being drivingly connected to an outupt shaft of said transmission; and
   - a selector mechanism arranged to selectively provide a drive connection between said second output shaft and said rear or front road wheels or a drive connection between said differential carrier and said rear or front road wheels.

2. A power transfer device as recited in claim **1**, wherein one of said side gears is formed smaller in diameter than the other side gear.

3. A power transfer device as recited in claim **1**, wherein said selector mechanism is arranged coaxially with said second output shaft.

4. A power transfer device as recited in claim **1**, wherein said selector mechanism is arranged in parallel with said second output shaft.

5. A power transfer device as recited in claim **1**, wherein said first output shaft is arranged to drive the front road wheels, and said second output shaft is arranged to drive the rear road wheels.

6. A power transfer device as recited in claim **1**, wherein said carrier is integrally connected with said differential case by means of a plurality of circumferentially equi-spaced support pins, and wherein said second differential includes a plurality of planetary gears each rotatably mounted on said support pins and in mesh with said sun gear and said ring gear.

7. A power transfer device as recited in claim **1**, wherein said differential carrier is arranged concentrically with said differential case, and said first and second output shafts are arranged in a fore-and-aft direction of the vehicle.

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